COMBUSTION

DEVOTED TO THE ADVANCEMENT OF STEAM PLANT DESIGN AND OPERATION

February, 1944

Photo by H. R. Towse

Three bowl-mill exhauster fans serving a 400,000 lb per hr boiler unit

Power Supply at Synthetic Rubber Plant ▶

Errors in Temperature Measurement by Radiometric Methods ▶

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FEATURE ARTICLES

EDITORIALS

DEPARTMENTS

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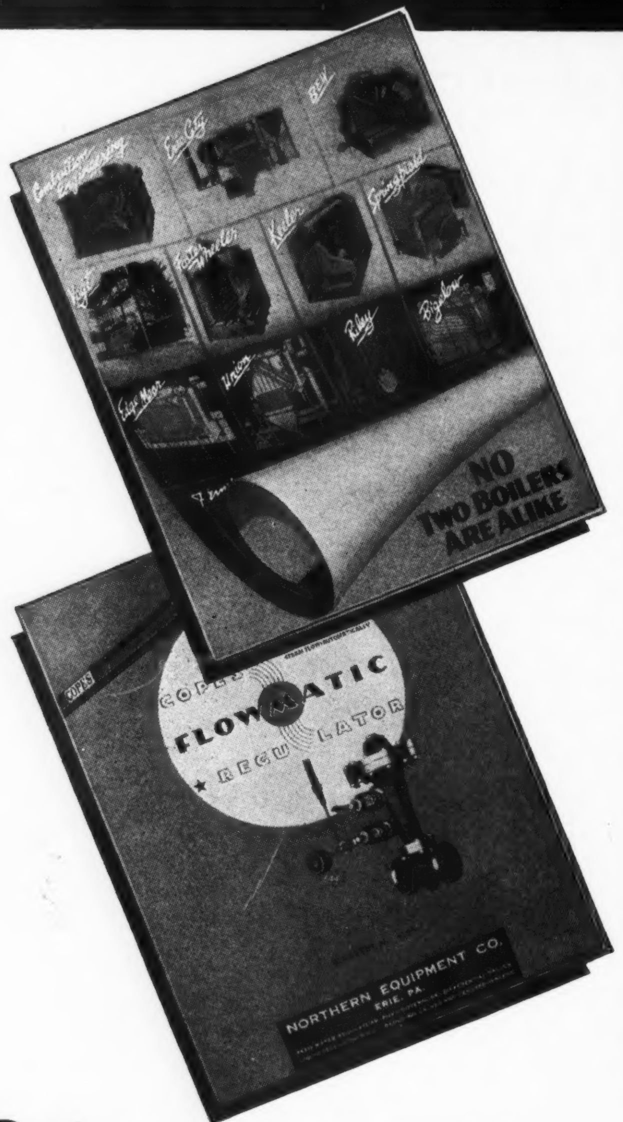

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EDITORIAL

Meeting the Post-War Oil Demand

Commenting on the world oil situation, *The Lamp*, a publication of the Standard Oil Company (N. J.), points out that our proved oil reserves are small compared with the probable demands of a world at peace, and that Russia alone, among first-rank nations, possesses oil reserves which appear to make her self-contained as to oil requirements over a long future. It observes that, although Imperial Russia was an exporter of oil, since the revolution Russian oil has been reserved almost entirely to meet the internal economy of that country. Furthermore, the view is expressed that this policy is likely to be continued and that Russian oil resources, large as they appear to be, probably will not be made available to the rest of the world.

The basis for this conclusion is not explained, but if such a policy should prevail it will prove a partial disillusionment to those who, looking ahead to post-war trade, are advocating the importation of raw materials to supplement our depleted reserves and as payment for the export of manufactured products from this country. Also, it will disappoint those who have hoped that some such means might be found for reciprocal payments under lend-lease for equipment for post-war use.

Our present proved reserves have been estimated as sufficient to meet peace-time requirements for about fifteen years at the immediate pre-war rate of consumption, but there are many who contend that more extensive wildcat drilling would have raised this figure considerably had not governmental price restrictions prevailed to dull the incentive. Nevertheless, the war has produced a tremendous drain on our oil resources which is likely to result in post-war conservation measures, in some form or another. Moreover, it is improbable that we will again be in a position, from the standpoint of national economy, to supply sixty per cent of the world's demand, as in the past. If a larger share of this demand were to be met by exports from Russia and the Near East, more oil would be available for meeting our domestic needs.

It is too early to speculate as to whether the recently announced plan of the Soviet Union, granting freedom of action to its component republics, will alter previous export policies concerning oil, but it is a matter that warrants close attention. If the principles set down in the Atlantic Charter are to apply to post-war policy between the nations, oil will be one of the most important raw materials to be made available on equal terms for world need. However, there is little assurance that these principles will take concrete form.

Significant were the recent comments of the President concerning our part in the pipe line to be built across Arabia from the Persian Gulf. These were to the effect that the United States must now look to foreign oil.

Collective Bargaining Within the A.S.C.E.

Widespread interest has been created by a vote of the American Society of Civil Engineers at its Annual Meeting in January, in sustaining previous action by the Board to permit local sections to set up collective-bargaining agencies for members of these respective groups. Moreover, the Society is reported as having appropriated fifty thousand dollars to engage four field representatives to handle the program.

Although the question of unionization for engineers has long been argued within the ranks of several of the societies, others have thus far chosen to adhere to their original professional aims without attempting to encompass the economic status of their members. The A.S.C.E is therefore the first of the national professional societies to depart from this policy and, being the oldest, its recent action is all the more startling to those accustomed to regard professional engineering as in a distinct category.

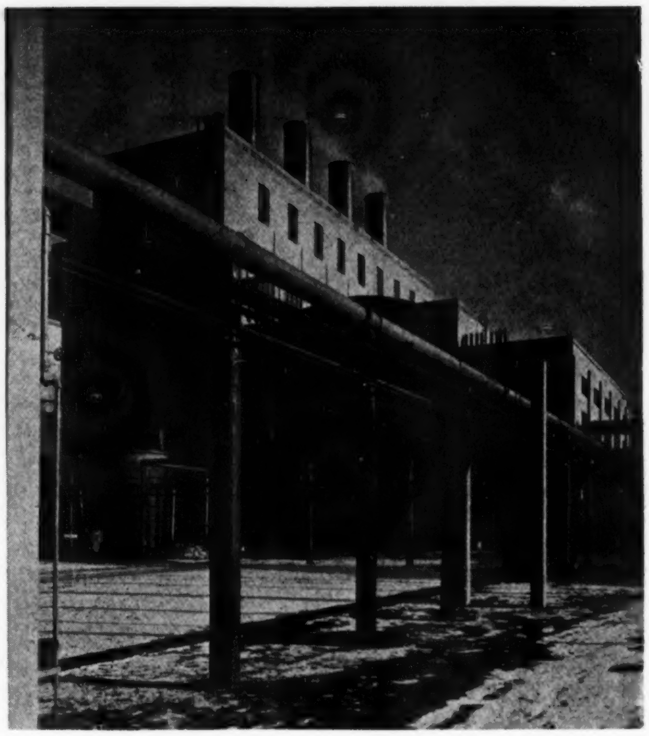
In defense of the step, it is contended that many of its members, by force of circumstances under the present labor movement, have been compelled to join various subprofessional unions affiliated with the A.F. of L. or the C.I.O., and that it is therefore preferable to provide means for collective bargaining by representatives of the engineers concerned.

It is undeniable that salaried men as a class have been squeezed between taxation and high living costs, on the one hand, and freezing of salaries under government decree, on the other. Engineers in particular, with a background of technical education and experience, are inclined to view the gains of organized labor under benevolent government influence as out of line with the qualifications involved, and it is not to be wondered that many of the junior engineers look to collective bargaining as the solution.

Nevertheless, unlike labor, the work of professional engineers cannot be measured in hours, seniority is not a measure of ability, renumeration cannot be equitably standardized because of the many factors involved, and promotion depends upon the individual's aptitude and ability to assume added responsibilities. Moreover, many engineering positions involve supervisory duties which tend to complicate the situation where unionization is concerned. While collective bargaining tends to raise the minimum level of compensation, it also tends, in the end, to hold down rewards for ability in the upper levels.

At present the action of the A.S.C.E. appears merely permissive, but such movements soon gain momentum and extend beyond the original concepts; they are likely to become restrictive and, as such, in conflict with the professional engineer's traditional individual freedom of action.

Power Supply at Synthetic Rubber Plant



Courtesy Engineering News-Record

Fig. 1—Exterior view shows size of the reinforced-concrete power house

Constructed entirely of reinforced concrete, this plant contains four 350,000-lb per hr steam generating units, three of which employ natural circulation and one forced circulation. These supply steam at 725 psi, 750 F to a 35,000-kw turbine-generator which exhausts to process at 165 psi. Excess electrical energy is supplied to the local utility. The plant produces butadiene and styrene.

SPEAKING before the Metropolitan Section, A.S.M.E., on January 18, John Saxe, vice president of Gibbs & Hill, Inc., described the new power plant of the synthetic rubber plant of the Koppers Company in western Pennsylvania. This furnishes steam and power for the production of butadiene and styrene, the installed capacity being 1,400,000 lb of steam per hour at 725 lb pressure and 750 F with four boilers and a single 35,000-kw turbine-generator exhausting at 165 psi to process. Selection of steam conditions was influenced by the high makeup required and one large, rather than two smaller turbine-generators, was due to conditions existing at the time in the turbine manufacturer's shops. Inasmuch as only about 13,000 kw of electric energy is required, the excess is supplied to a local utility to meet outside requirements.

Mr. Saxe's talk was supplemented by remarks from W. A. Armacost, vice president of Combustion Engineering Company; N. J. Connor of Babcock & Wilcox Company and C. C. Franck of Westinghouse Company, each of whom dealt with details of the equipment furnished by their respective companies.

Because of the critical steel situation at the time the plant was laid down, reinforced concrete construction was adopted throughout for the building except for steel used in carrying the boilers and for stacks. None was employed for lateral bracing and an 80-ft concrete arch spans the turbine room. By using reinforced concrete some 1200 tons of steel was saved, although the building cost was thereby considerably increased. It is believed to be the largest reinforced-concrete power plant yet constructed. Ground was broken in June 1942 and the first boiler was started a year later, some delay having been encountered in the concrete work during the winter because of bad weather.

The power plant contains four steam-generating units, each rated at 350,000 lb per hr continuous output. Their maximum size was dictated by available draft equipment and geared drive for the fan turbines. Three of these boilers employ natural circulation and the fourth forced circulation. The fuel is pulverized coal and all have dry-bottom furnaces.

Because practically all the steam is used in the process except that required by auxiliaries, the makeup is very high, as is also the blowdown. Feedwater is taken from the Ohio River and, after screening, is treated in two stages by cold lime-soda process softening and carbonaceous zeolite. Boiler water concentrations are high, ranging from 2500 ppm to 4500 ppm. Feedwater is heated in two stages, namely, from 140 F to 240 F in open deaerating heaters which take steam from the auxiliary turbine exhaust line; then to 362 F in closed heaters which receive steam from the main turbine exhaust and from the blowdown flash tank.

All piping in the plant is welded and the welds were stress-relieved by the induction method. The forced- and induced-draft fans are turbine-driven through reduction gears and the combustion control is of the all-electric type. The coal-handling equipment has a capacity of 400 tons per hour and 30-hr storage is provided. All ash from the hopper-bottom furnaces is continuously sluiced to fill.

Steam-Generating Units

The three B. & W. natural-circulation units are of the two-drum radiant type without convection surface, as shown in the cross-section, Fig. 2. In fact, screen tubes in front of the superheater have been omitted as the calculated furnace heat release at full rated load of 350,000 lb per hr is only 14,100 Btu per cu ft. An average of 65,000 Btu per sq ft of heat-absorbing surface is available. The rear and side-wall tubes are $2^1/_2$ in. and those on the burner wall $3^1/_4$ in. with stud plates welded on between rows. Air preheaters of the regenerative type are provided but there are no economizers.

Each unit is served by two Hardinge mills, with each mill supplying three burners of the horizontal circular type. Two burners on one boiler are arranged to burn casing-head gas if desired.

The forced-circulation unit is shown in Fig. 3. This has a single 54-in. drum 20 ft long and the furnace wall tubes, which are plain, are $1^1/_4$ in. on $1^5/_{16}$-in. centers, trifurcated so that three tubes merge into a single tube entering the bottom header. At the top the tubes enter the header in staggered ligaments. Thus the number of distributing nozzles in the bottom header is reduced. The tubes are welded to the bottom header and rolled into the top header. The superheater is shielded by two rows of wall tubes.

Referring to the cross-section, the Elesco economizer is in two sections and the surface shown below the economizer is secondary evaporating surface in the forced-circulation circuit. This was necessitated by the fact that the economizer does not heat the water up to full saturation temperature. In this section, which is between the economizer and the drum, the ratio of water circulated to steam produced is 4 to 1, whereas in the furnace-wall circuits it is about $5^1/_2$ to 1. Circulation is provided by two pumps and is independent of the load on the unit. The calculated furnace heat release was 18,200 Btu per cu ft.

Two C-E Raymond bowl mills supply pulverized coal to four horizontal burners (two per mill). A regenerative air preheater of the Ljungstrom type supplies hot air at around 390 F.

According to Mr. Armacost, the forced-circulation unit effected a considerable saving in weight as compared with a natural circulation unit of like capacity; this comparison was of the order of $151^1/_2$ tons to 237 tons. In commenting on operation of the forced-circulation unit, he observed that when operating with 2500 ppm boiler-water concentration, the load has been thrown on and off five times in 15 min with resulting nearly constant water level. Subsequent to making some alterations in the drum internals, it has been possible to carry a load of about 400,000 lb per hr with steam showing only 1 ppm solids.

Turbine-Generator

This unit, of special design, has one Rateau stage and nine reaction stages and is designed for a maximum flow of 1,277,000 lb of steam per hr. The blades are shrouded and, although rated at 35,000 kw, provision is made for bypassing the first, or Rateau, stage at which time it is capable of producing 45,000 kw, or 5000 kw per stage. While it exhausts at 165 lb, steam can be extracted at 400 lb. There is a reducing valve and desuperheater by-pass for supplying steam direct to process should the turbine be down. The turbine is provided with two 16-in. throttle valves, one on each side, and the whole unit is of particularly sturdy construction.

Although the plant is designed for non-condensing operation, space has been provided for a condenser if future conditions make its installation desirable.

Simplicity and reliability in operation motivated the plant layout and to this end conservative values were adopted in the design of its components. Operation thus far appears to confirm the wisdom of this course.

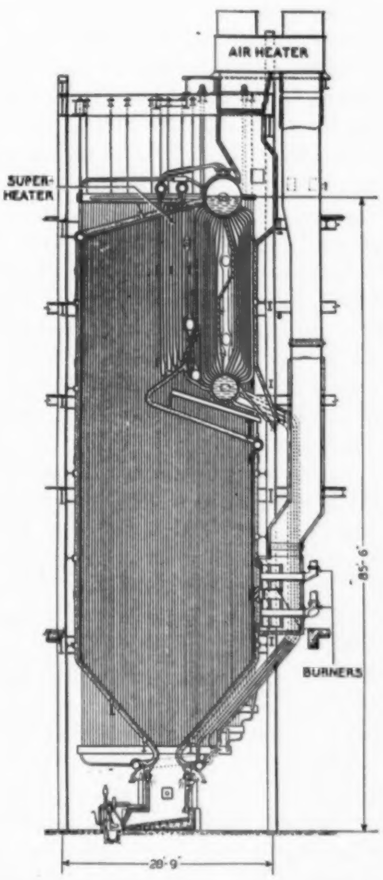


Fig. 2—B. & W. natural circulation boiler

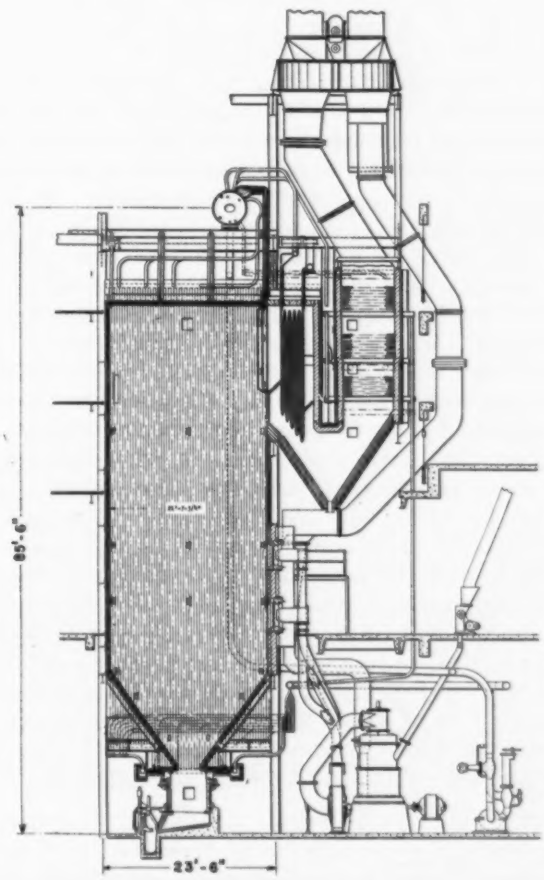


Fig. 3—C. E. controlled forced-circulation unit

Errors in Temperature Measurement by Radiometric Methods*

By W. T. REID† and R. C. COREY‡

Corrections for using optical and total-radiation pyrometers under imperfect conditions have been available for the case where the object whose temperature is being determined is hotter than its surroundings. However, if the object is a cooled surface surrounded by a higher temperature environment, such as furnace wall tubes, different corrections must be applied. The fundamental reasons for this difference are here discussed, methods used in calculating corrections are given, and tables of corrections are included. If the temperature to be measured is higher than that of its surroundings, the optical pyrometer is less affected by changes in emissivity of the radiating surface than is the total radiation pyrometer, and it is the reasonable instrument to use where a recording instrument is not required. However, where temperature of the surroundings is higher than that of the object whose temperature is sought, the radiation pyrometer is more precise.

During an investigation of the physical and chemical properties of certain materials found externally on the wall tubes of slag-tap, pulverized-coal furnaces, it was desirable to determine the surface temperature of various surfaces in an experimental furnace designed to simulate actual furnace conditions. These surfaces consisted of water-cooled steel specimens coated with oxide and with various salts. Although thermocouples placed in the metal specimens gave accurate values for the actual metal temperature, it was impossible to determine from these data the surface temperature of oxides or deposited materials because of lack of information on their heat-transfer coefficients. Because of the nature of the material on the surface of the metal and its small dimensions, thermocouples could not be applied directly, and resort was made to radiation methods for temperature measurement. However, actual furnaces are never perfect black-body radiators, and even carefully designed laboratory equipment seldom reaches ideal radiation conditions; therefore, corrections must be applied to radiometric measurements to compensate for errors introduced by this factor.

Published information indicates that such corrections have been in common use for the case where the object being measured is at a higher temperature than its surroundings, but, as in boiler furnaces, when the object is at a lower temperature than the surroundings, similar corrections have not been available. Although the magnitude of these corrections will be obvious to the physicist or to the specialist in temperature measurement, many users of total radiation or optical pyrometers may not be aware of the great difference existing between the corrections to be applied under these different conditions, and it is believed that a brief discussion of this subject would be of value. This article, therefore, is concerned with the corrections to be applied to temperature measurements obtained by radiometric methods of objects cooler than their surroundings, water-cooled walls in pulverized-coal furnaces being a notable example.

Fundamentals of Radiometric Pyrometry

Emissivity.—Determination of the temperature of an object by either total radiation or optical pyrometers depends on measurement of the intensity of the radiant energy emitted from the object, but this intensity of radiation depends not only on the temperature of the object but also on the nature of its surface. The ability of surfaces to emit or absorb radiant energy varies widely between different materials, being a maximum for such surfaces as platinum black or carbon black and a minimum for bright surfaces, such as polished platinum or a silvered mirror. Thus, when radiant energy falls on a body, absorption, reflection or transmission of the energy may occur, depending on the nature of the body, and in some instances all three phenomena may occur simultaneously. For metals and most other opaque materials, only absorption and reflection are involved, and where all of the energy is absorbed and none is reflected, the object may be defined as a black body.

Such conditions can be realized experimentally by heating an enclosure, such as a spherical shell, to uniform temperatures and observing the radiation coming from a small hole in the wall. The effectiveness of a surface as an emitter or absorber of energy is measured in terms of *emissivity*, which is defined as the ratio of the energy emitted from a given body at a certain temperature to that emitted by a black body at the same temperature—a black body having an emissivity equal to 1.0. In other words, a black body will radiate more energy than any other object at the same temperature.

A relationship important in correcting observed temperature readings made under other than black-body conditions is that $E + R = 1$, where E is the emissivity and R is the reflection coefficient for the surface. Thus, for a mirror having a *reflection* coefficient of 0.95, the emissivity is only 0.05; whereas for a substance like carbon black, having an *emissivity* of 0.98, only 0.02

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* Published by permission of the Director, Bureau of Mines, United States Department of the Interior.
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of the energy falling on the surface is lost by reflection. It is this relationship that is used below in conjunction with other fundamental laws in computing the corrections to be applied to observed readings made under non-black-body conditions.

Total Radiation Pyrometers.—These instruments, usually consisting of a multiple-junction thermocouple of small dimensions on which is focused the radiation from the object being measured, generate a small voltage because of the rise in temperature of the hot junction. The extent of this temperature rise is nearly proportional to the rate at which radiant energy (that is, the intensity of radiation at *all* wave lengths) is received by the instrument; and if the effect of the temperature of the receiver and absorption by lenses or other focusing devices is assumed to be negligible, it is proportional to the fourth power of the absolute temperature of the radiator; actually, the response may be somewhat different. This relationship is based upon the Stefan-Boltzman law, which, in simplified form, is

$$J = cT^4 \tag{1}$$

where J is the total energy radiated per unit time over unit area and c is a constant. For non-black-body conditions, it becomes:

$$J' = cET^4 \tag{2}$$

where E is the emissivity of the radiator.

Optical Pyrometers.—Fundamentally, these instruments are photometers that compare the intensity of radiation from the radiator at a fixed wave length with that from a known source, such as the filament of an incandescent lamp. The comparison is made at a single wave length by viewing the two fields through a monochromatic filter, generally red. Inasmuch as only the energy comprising a narrow band of wave lengths is transmitted by the filter, and the intensity of radiation varies with the wave length, the temperature scale of optical pyrometers is based upon the Wien law, which gives the relation between radiant energy and wave length for a black body as:

$$J_\lambda = c_1\lambda^{-5}e^{-\frac{c_2}{\lambda T}} \tag{3}$$

where

J_λ = intensity of radiation per unit area at wave length λ
c_1 = first radiation constant[1]
c_2 = second radiation constant
T = absolute temperature of the black body
e = base of Napierian logarithms

For non-black-body conditions:

$$J_\lambda' = c_1E_\lambda\lambda^{-5}e^{-\frac{c_2}{\lambda T}} \tag{4}$$

where E_λ is the emissivity of the surface at wave length λ.

Experiments have shown that these equations are accurate to within one per cent for temperatures as high as 9000 F.[2] From these equations, and certain modifications of them, it is possible to calculate the corrections to be applied to any observed temperatures determined by radiometric methods if the emissivity of the surface is known for the given conditions. In addition, for the case where the surroundings are at a higher temperature than the radiator, additional corrections must be applied, as shown later.

[1] Wensel, H. T., "International Temperature Scale and Some Related Physical Constants," National Bureau of Standards *Journal of Research*, vol. 22, April 1939, p. 375

[2] Forsythe, W. E., "Measurement of Radiant Energy," McGraw-Hill Book Co., New York, N. Y., 1936, p. 367.

A form of optical pyrometer has been suggested in which the effect of emissivity can be canceled by making observations through different-color filters, usually red and green. Thus, the intensity of radiation at two different wave lengths is determined, from which the true temperature of the surface can be calculated if it is assumed that there is no change in emissivity with a change in wave length. Such two-color instruments are not commercially available; difficulty in calibrating them is great, and more precision is usually required than is available in portable instruments.

Corrections for Hot Objects Viewed in the Open

As noted previously, corrections to be applied for an emissivity of less than 1.0 of objects viewed in the open, such, for instance, as a steel ingot while cooling or slag while being tapped from a furnace, have been computed already by many investigators and are to be found in most standard texts on temperature measurement. For optical pyrometers, the corrections may be computed from an equation derived from Wien's law, of the form:

$$\frac{1}{T} - \frac{1}{S} = \frac{\lambda \log_e E_\lambda}{c_2} \tag{5}$$

where

T = actual absolute temperature, in degrees Kelvin
S = observed absolute temperature, in degrees Kelvin
λ = effective wave length transmitted by pyrometer filter, in microns
E_λ = emissivity of object at wave length λ
c_2 = second radiation constant, 14,360 centimeter-degrees $\times 10^{-4}$

For the conventional filter of $\lambda = 0.65\ \mu$, this equation simplifies to

$$\frac{1}{T} - \frac{1}{S} = \frac{\log_{10} E_\lambda}{9593} \tag{6}$$

Corrections computed by means of this equation by Foote and others[3] are given in Table 1 for three common materials. The effect of the low emissivity of molten

TABLE 1—OBSERVED AND TRUE TEMPERATURES MEASURED BY OPTICAL PYROMETERS USING RED LIGHT (λ = 0.65) WHEN SIGHTED UPON THE FOLLOWING MATERIALS IN THE OPEN

Observed Temperature, °F	True Temperature, °F		
	Molten Cast Iron*	Solid Iron Oxide	Molten Slag†
1400	..	1400	..
1600	..	1603	..
1800	..	1806	..
2000	2148	2011	..
2200	2374	2218	..
2400	2602	..	..
2600	2831	..	2702
2800	3063	..	2917
3000	3299	..	3135

* For E_λ = 0.40.
† For E_λ = 0.65.

iron causes a large correction in the observed temperature, whereas iron oxide, having a high emissivity, is affected but little. Although molten iron is usually assumed to have an emissivity of 0.40, some recent work[4] has shown that it may range from 0.40 to 0.75, depending upon the analysis. Alloying elements such as Cr and Mn increase and Ni and Si decrease the emissivity. As an example of the variation in apparent temperature to be expected from such changes, a plain carbon steel with a true temperature of 2860 F, as measured by means of a thermocouple, gave an optical pyrometer reading of

[3] Foote, P. D., Fairchild, C. O., and Harrison, T. R., "Pyrometric Practice," Bur. Stds. Tech. Paper 170, 1921, 326 pp.

[4] Goller, G. N., "The Emissivity of Molten Stainless Steels," A.S.M. Preprint No. 28, 1943.

2650 F. A 5 per cent Cr steel at the same true temperature showed an optical pyrometer reading of 2710 F.

Similar corrections to those of Table 1 have been computed by the same authors for a total radiation pyrometer used under the same conditions; these corrections are shown in Table 2.

TABLE 2—OBSERVED AND TRUE TEMPERATURES MEASURED BY TOTAL RADIATION PYROMETER WHEN SIGHTED UPON THE FOLLOWING MATERIALS IN THE OPEN

Observed Temperature, °F	True Temperature, °F	
	Molten Iron	Iron Oxide
1400	2090	1469
1600	2370	1678
1800	2645	1888
2000	2920	2098
2200	3185	2308

Obviously, for objects viewed in the open and having a low emissivity, such as molten iron, the total radiation pyrometer is less suitable than the optical pyrometer because of the greater variation in observed temperature with change in emissivity. Nevertheless, the radiation pyrometer is used successfully for control purposes under such conditions when objects of nearly constant emissivity are being viewed from day to day and it is desirable to record their apparent temperature, as with a recording potentiometer. With objects having an emissivity of less than 1.0, that is, objects that are not perfect radiators, temperature measurements made with an optical pyrometer will be less in error than those made with a total radiation pyrometer, when all other conditions are equal and the object is viewed in the open, that is, when it is surrounded by an environment of much lower temperature.

Corrections for Objects Surrounded by a Hotter Environment

Although the previous discussion is based upon a wide field of utilization of radiometric methods, the corrections of observed temperatures already shown are not applicable to the condition where the object whose temperature is being determined is surrounded by an environment of higher temperature. Tables of corrections for such conditions are not ordinarily available, probably because such corrections are affected not only by the temperature and emissivity of the object, but also by the temperature of the surroundings. Methods of calculating these corrections and tables based upon conditions ordinarily encountered in practice will be given later.

Many industrial furnaces, such as are used for heat treating or annealing, approach black-body conditions closely when at equilibrium; and objects in the furnace, being at the same apparent temperature as the furnace walls, cannot be distinguished from the background. In the case of modern boiler furnaces having water-cooled walls, however, black-body conditions are never attained, and broad assumptions are required to calculate the effect of surroundings upon apparent temperatures. For simplification, the case to be described will refer to a refractory-lined furnace suitably enclosed, so that by itself it represents a black body with a water-cooled steel surface set into one wall having negligible area as compared to the area of the furnace walls and maintained at a fixed temperature lower than the temperature of the furnace. What will be the apparent temperature of the surface of this object under varying conditions of surface and of actual temperature?

Corrections With Optical Pyrometer.—As shown previously, different corrections may be expected by reason of surfaces having different emissivities. For instance, if a water-cooled first-surface mirror was installed inside the furnace and maintained at 100 F by an ample flow of water, an optical pyrometer sighted upon it would indicate a surface temperature approximating that of the furnace walls, because the reflectivity of the mirror is high for the radiation at a wave length of 0.65 μ. However, if the surface of the mirror was obscured, as by a layer of soot deposited from an acetylene flame, it would appear to have a much lower temperature than the furnace wall, although in both of these instances its true temperature has not changed.

This suggests for these conditions that the reflectance from the surface is of primary importance in fixing the apparent surface temperature, remembering, however, that the reflection coefficient depends upon the emissivity according to the relationship $E + R = 1$. Thus, an object having an emissivity of less than 1.0 will emit energy according to the Wien distribution law corrected for non-black-body conditions and, in addition, will reflect energy received from the surroundings. When these surroundings are at a low temperature, such as at ordinary room temperature, obviously no energy will be reflected, and the simple corrections tabulated in Table 3 for the effect of change in emissivity can be applied directly. When the surroundings are at the same temperature as the object, black-body conditions are attained and no corrections need be made. When the surroundings are at a higher temperature than the object, the latter still emits energy, dependent on its temperature and emissivity, but at the same time energy from the surroundings is partly reflected from the surface of the object and increases its *apparent* surface temperature.

Corrections for this latter condition, when optical pyrometers are employed, are most easily computed by using a modification of equation (5), as noted by Burgess and LeChatelier,[5] by means of which relative intensities of radiation can be compared. The equation is:

$$\log_{10}\frac{J}{J_1} = \frac{c_2 \log_{10} e}{\lambda}\left(\frac{1}{T_1} - \frac{1}{T}\right) \qquad (6)$$

using the same nomenclature as before, where J is the intensity of radiation at temperature T, and J_1 the intensity at T_1. A typical calculation using this equation follows:

Assume a furnace temperature of 2700 F, with the water-cooled metal surface maintained at room temperature and having an emissivity of 0.95. What is its apparent surface temperature when observed with an optical pyrometer?

First, $E = 0.95$, consequently $R = 1 - E = 0.05$.

Thus, if J is the intensity of radiation at 2700 F and is taken as 1.0, then J_1 will be the intensity of radiation reflected from the cool surface and will be 0.05. Because at room temperature the energy actually being emitted from the surface is negligible, it can be neglected in these calculations. Thus, the problem is one of finding the apparent temperature due to the energy being reflected from the cool surface.

Substituting in equation (6) and converting 2700 F to degrees Kelvin,

[5] Burgess, G. K., and LeChatelier, H., "The Measurement o fHigh Temperatures," John Wiley & Sons, New York, p. 254, 1912, 510 pp.

$$2700\text{ F} = 1482\text{ C} = 1755\text{ K}$$

$$\log \frac{1}{0.05} = \frac{(14{,}360)(0.4343)}{0.65}\left(\frac{1}{T_1} - \frac{1}{1755}\right)$$

$$\frac{1}{T_1} = 0.0007054$$

$$T_1 = 1418\text{ K or }2093\text{ F}$$

Therefore, the apparent surface temperature of the specimen would be 2093 F.

Similar calculations made for different assumed emissivities, but with the other conditions the same, are summarized in Table 3. Thus, even with an emissivity

TABLE 3—EFFECT OF CHANGE IN EMISSIVITY ON APPARENT TEMPERATURE OF A SURFACE ACTUALLY AT ROOM TEMPERATURE WHEN SURROUNDED BY AN ENVIRONMENT AT 2700 F, AS DETERMINED BY AN OPTICAL PYROMETER

Emissivity	Apparent Surface Temperature, F.
0.10	2674
0.50	2534
0.90	2210
0.95	2093
0.99	1852
0.999	1578

as high as 0.999, where only 0.1 per cent of the incident energy is reflected, the apparent temperature of a body at room temperature in an environment of 2700 F would be 1578 F if measured with an optical pyrometer.

In an actual experiment in this laboratory, a water-cooled steel specimen provided with several thermocouples to measure the rate of heat transfer and the temperature of the metal at the heated surface was installed in a gas-fired furnace. With the furnace at a temperature of 2700 F a layer of oxide gradually formed on the surface of the specimen, which then appeared to be at 2210 F by measurement with an optical pyrometer. The specimen was removed, coated with a layer of soot from an acetylene flame, and replaced. Its apparent temperature then was only 1980 F, although the actual metal temperature and the rate of heat transfer were found to be unchanged. Thus, by increasing the surface emissivity, apparent temperature was decreased 230 deg F.

It is obvious that under these conditions the optical pyrometer is useless for determining true surface temperature. The great effect of a small change in emissivity is clearly shown, and it is impossible to attain emissivities high enough to permit reasonable corrections.

A similar method of calculation can be used to predict the apparent temperature of the surface when it is maintained at temperatures higher than room temperature but below that of its environment. In this instance it is necessary to make an allowance for the intensity of radiation actually emitted by the specimen. Thus, for the same conditions as in the sample calculation given above, but assuming that the *true* temperature of the surface is 2093 F, and therefore it emits 0.05 of the energy at 2700 F, the energy received by the optical pyrometer would be 0.95 times this energy emitted from the surface at a temperature of 2093 F plus the reflected energy, which itself amounts to 0.05 times that at 2700 F. Therefore, the total intensity of radiation from the surface of the specimen would be $(0.95)(0.05) + 0.05 = 0.0975$ of that emitted from a surface at 2700 F.

Consequently, again using equation (6):

$$\log \frac{1}{0.0975} = (9595)\left(\frac{1}{T_1} - \frac{1}{1755}\right)$$

$$\frac{1}{T_1} = 0.000675$$

$$T_1 = 1481\text{ K or }2206\text{ F}$$

and the apparent surface temperature is 113 deg F higher than the true temperature under these conditions.

Similar computations have been made for various temperatures of the surface and for different emissivities. Table 4 shows a comparison between actual and observed temperatures based upon these calculations. As noted

TABLE 4—TRUE AND APPARENT TEMPERATURES MEASURED BY AN OPTICAL PYROMETER WHEN SIGHTED ON A SURFACE SURROUNDED BY AN ENVIRONMENT AT 2700 F

True Temperature of Surface, F	Apparent Temperature, F: $E_\lambda = 0.50$	$E_\lambda = 0.95$	$E_\lambda = 0.99$
100	2534	2091	1852
1000	2534	2091	1852
1500	2535	2092	1858
2000	2540	2163	2046
2500	2618	2516	2505
2600	2655	2609	2604

before for these conditions, even with an emissivity as high as 0.99, which is difficult to obtain for any surface, the optical pyrometer cannot be used as a measure of temperature because it cannot distinguish between radiation reflected from the surface and radiation being emitted from the surface. As has been shown, when the temperature of the surface approaches that of the environment, the corrections decrease. This suggests the successful use of the optical pyrometer under these conditions when the emissivity is high and the magnitude of the unknown temperature can be estimated approximately.

Corrections for the Total Radiation Pyrometer

Comparison of the Stefan-Boltzmann equation (eq 1), on which total radiation pyrometry is based, and the Wien equation (eq 3) shows that the energy change for the same temperature interval should differ; the former varies as the fourth power of the absolute temperature interval and the latter as a complex exponential function. The essential difference between the two methods is that one integrates the energy emitted at *all* wave lengths and the other measures it at a *single* wave length. Although the total energy received from a black body at any given temperature is greater with the total radiation pyrometer than with the optical pyrometer, the relative increment over a range of temperatures is much greater with the optical pyrometer. These relations may be seen in Table 5, the data of which were taken from the International Critical Tables.[6]

TABLE 5—COMPARISON OF TEMPERATURE-ENERGY RELATIONSHIPS FOR TOTAL RADIATION AND OPTICAL PYROMETERS

Temperature C	Temperature F	Total Radiant Intensity: $J = \epsilon T^4$ (Ergs/Sec/Cm²) $\times 10^7$	Total Radiant Intensity: Proportionate Increase in Intensity in 200 C Interval	Monochromatic Radiant Intensity*: $J\lambda = c_1\lambda^{-5}\epsilon^{-\frac{c_2}{\lambda T}}$ (Ergs/Sec/Cm³) $\times 10^4$	Monochromatic Radiant Intensity*: Proportionate Increase in Intensity in 200 C Interval
800	1472	7.6	..	3.4	...
			1.97		250.0
1000	1832	15.0		850	
			1.80		39.5
1200	2192	27.0		33,600	
			1.67		13.7
1400	2552	45.0		461,000	
			1.56		7.2
1600	2912	70.0		3,310,000	
			1.51		4.6
1800	3272	105.4		15,300,000	
			1.45		3.4
2000	3632	152.4	..	52,100,000	...

* Based on $\lambda = 0.65\ \mu$.

It is apparent that the increase in radiant intensity at the fixed wave length of the optical pyrometer is much greater for a given temperature change than is the increase in total radiation. This suggests that the corrections to be applied in using the total radiation py-

[6] International Critical Tables, vol. 5, p. 238, McGraw-Hill Book Co., 1929.

rometer may be less than when the optical pyrometer is used under conditions where reflectance of energy from the surroundings can occur from the surface observed.

Calculations also have been made to show the magnitude of these corrections when the total radiation pyrometer is used. Table 6 shows the comparison between true and apparent temperatures for surfaces of various emissivities when the surrounding environment is at a higher temperature than the surface. These data were calculated from the following relationships:

$$J_T = J_R + J_E = 0.173\left[(1 - E)\left(\frac{T_f}{100}\right)^4 + E\left(\frac{T_t}{100}\right)^4\right] = 0.173\left(\frac{T_i}{100}\right)^4$$

whence

$$\left(\frac{T_i}{100}\right)^4 = \left[(1 - E)\left(\frac{T_f}{100}\right)^4 + E\left(\frac{T_t}{100}\right)^4\right]$$

where

J_T = total energy received from the surface by the total radiation pyrometer
J_R = energy due to reflection
J_E = energy due to emission at the true temperature
T_t = true temperature of the surface, degrees absolute
T_f = temperature of surroundings
T_i = temperature indicated by the pyrometer
E = emissivity

TABLE 6—TRUE AND APPARENT TEMPERATURES MEASURED BY A TOTAL RADIATION PYROMETER WHEN SIGHTED ON A SURFACE SURROUNDED BY AN ENVIRONMENT AT 2700 F

True Temperature of Surface, F	Apparent Temperature by Total Radiation Pyrometer $E = 0.50$	$E = 0.95$	$E = 0.99$
100	2199	1041	563
1000	2228	1286	1071
1500	2291	1629	1528
2000	2413	2050	2010
2600	2651	2606	2602

Comparison of Tables 4 and 6 shows that the error for various emissivities is lower in each instance for the total radiation pyrometer, the difference being most pronounced where the true temperature is low and the emissivity of the surface is high. Thus, it is apparent that if the temperature of an oxidized-steel surface having an emissivity of 0.95 and an actual temperature of 1000 F were to be measured with a total radiation pyrometer when the environment was at 2700 F, the error would be about 285 deg F. Under the same conditions, an optical pyrometer would be in error by over 1000 deg F.

Conclusions

Temperature measurements by radiation methods are useful in many applications, particularly when the object whose temperature is being determined is inaccessible or of small size. Determinations by these methods, however, can be much in error unless proper corrections are applied to compensate for fundamental differences in imposed conditions and the nature of the surface.

In the case of hot objects surrounded by a cold environment, such as steel ingots viewed in the open, the optical pyrometer is less affected by the emissivity of the surface than is a total radiation pyrometer and consequently gives a better indication of temperature when the nature of the surface cannot be determined directly.

When it is desired to determine the temperature of an object surrounded by an environment of higher temperature, such as the water-cooled walls of a pulverized-coal-fired furnace, total radiation pyrometers offer a more precise method of measurement than do optical pyrometers, because the nature of the surface emitting radiant energy is less important.

Addition to Capacity and Modernization of

MARYSVILLE POWER HOUSE—III

By H. E. MACOMBER
The Detroit Edison Company

The first two articles of this series, one in the October and the other in the December 1943 issue, described the new addition to the original plant built to accommodate part of the new equipment involved in the present increase in capacity. Additional space required for the rest of this equipment was provided by removing some of the old equipment in that end of the plant adjacent to the new addition. According to the general plan, when further increases in capacity are desired, more of the old equipment will be removed to make room for additional new equipment duplicating that installed in the initial step. The previous articles also described general design considerations, foundation problems, the new steam generators and the design and arrangement of main and auxiliary electrical systems. This, the third of the series of articles, covers coal handling, ash collecting and handling, turbine-generators and heat balance.

Coal Handling

IT WAS brought out earlier that the original boiler house contained two rows of double-end stoker-fired boilers served by a center and two side rows of coal bunkers. Each center row bunker, of 500-ton capacity, served a battery of two boilers on opposite sides of the main aisle. Also, each boiler was served by one of the 250-ton capacity side row bunkers. The bunker capacity associated with each boiler was therefore 500 tons. In selecting the new steam generators to be used in rebuilding the plant, the size was so chosen that new units could be accommodated in the positions occupied by the old boilers without changing the column spacing. To provide a unit of sufficient capacity, however, necessitated removal of the side row bunkers. Thus, in the ultimate boiler house, when all old boilers might be replaced with new steam generators, the only existing bunkers to remain would be the center row bunkers which would be insufficient.

Two of the three new steam generators involved in the first step, with which these articles are concerned, were installed in the extension of the boiler house; the third was installed in a position previously occupied by one of the eight old boilers. In the extension, a 500-ton bunker was installed between the two new units to match the center row bunkers in the old boiler house.

In making provisions for the additional active storage capacity required by the three new steam generators, it was found to be economical from the long range point of view to provide sufficient capacity of this kind to serve the completely rebuilt boiler house; that is, to serve the total of ten new steam generators that may ultimately occupy the boiler house. By active storage is meant, storage within the plant from which coal can be fed to the steam generators by the operating personnel without the aid of yard-handling equipment or crew.

In determining the active coal storage that should be provided for a plant, several items must be taken into consideration; for example, the number of hours per day and the number of days per week the yard equipment is to be in service; the maximum coal requirement of the plant during the longest period the yard equipment is to be shut down; decreased capacity of coal-handling equipment due to wet or frozen coal, allowances for maintenance, breakdown, etc. Taking all such factors into account, studies showed that for the ultimate Marysville Plant the maximum active coal storage should be of the order of 3000 tons. Although the ultimate plant would contain five 500-ton bunkers, or a total of 2500 tons capacity, it would not be practicable to arrange matters so that all bunkers could be completely filled when the yard equipment is shut down, nor would it be feasible to count on drawing nearly all coal from the bunkers before the yard equipment is started again. Consequently, in determining the additional active storage capacity required, the bunker capacity was taken as half the maximum bunker capacity, or 1250 tons.

To provide the additional capacity, a steel silo 80 ft high, 35 ft in diameter at the top and 38 ft in diameter at the bottom, was built in the end of the new extension of the boiler house. The purpose of converging the walls of the silo from bottom to top was, of course, to avoid packing and bridging of the coal in its downward movement. This silo has a capacity of about 2000 tons giving a total active storage capacity for the ultimate plant of about 3250 tons. Meanwhile, of course, the remaining stoker-fired boilers will continue to be served by both the center row and side row bunkers.

An isometric view of the remodeled coal-handling system is shown in Fig. 27. The coal crushers are located

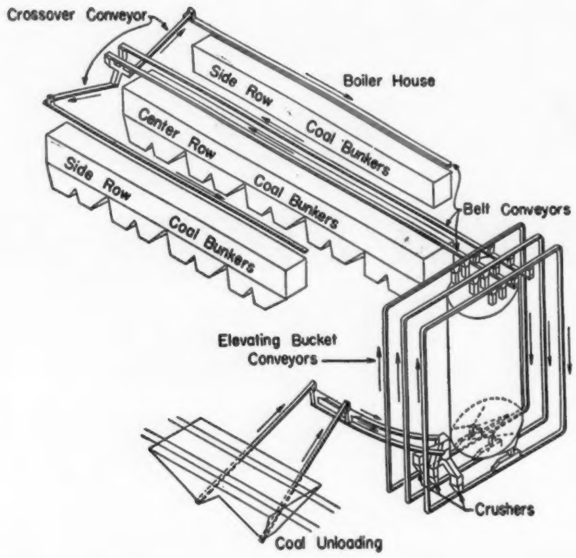


Fig. 27—Isometric view of remodeled coal-handling system

near the base of the silo. They are served by three bucket conveyor systems which, at the top of the boiler house, dump the crushed coal either into the silo or onto either one of the two conveyor belts that run the length of the boiler house over the center row bunkers. Coal for the side row bunkers, as before, is transferred to their respective belt conveyors by means of the crossover conveyors at the end of the boiler house. The three bucket conveyors mentioned above are also used for conveying coal from the silo to the conveyor belts that supply the bunkers. This operation is carried on as a routine procedure by the operating personnel during the period when the yard equipment is shut down.

Nearly all coal used at the Marysville Plant is delivered by self-unloading barges which deposit their cargo on a dock storage pile. From this pile the coal is loaded into cars by means of a drag-line scraper and ramp and carried either directly to the pits for delivery to the crushers or to the yard for storage. The navigation season covers a period from about April 1 to December 1. It is desirable, therefore, that at the close of the season, yard storage be sufficient to supply the plant needs until the reopening of navigation in the spring. There is space at the plant for a normal yard storage of about 250,000 tons.

Sampling of Coal

The sampling of coal consumed by a large steam-generating plant, in order to determine accurately the heat chargeable to the plant for purposes of computing thermal performance, is an important function. At Marysville provision has been made for automatically sampling the coal by diversion of the contents of one of the buckets in each of the three bucket conveyors to a sample bin each time the bucket makes a complete cycle of the loop when transporting coal from the crushers. When the conveyors are transporting coal from the silo to the bunkers, the resampling of the coal is avoided by means of electrical interlocks which prevent the sampling buckets from discharging into the sample bin. Two of the three bucket conveyors are of the same capacity and the third is of somewhat larger capacity. In order that the coal sampled from each conveyor be proportionately the same, a 50-lb capacity sample bucket is used on each of the two smaller capacity conveyors, and a 60-lb capacity sample bucket is used on the third conveyor. These capacities are such that the coal diverted to the sample bin will be of the order of 0.1 per cent of all the coal entering the plant.

It can be seen in Fig. 28 that coal dumped by the sampling buckets falls by gravity into a sample storage bin of some 9500-lb capacity. This bin is air-tight and heat-insulated to prevent condensation of moisture within. From the sample storage bin the coal is passed periodically, usually once a day, to a sample crusher mill. The mill is of 1 ton per hour capacity and reduces the coal to a fineness of 95 per cent through a four-mesh sieve. Leaving the mill, the coal falls by gravity through riffle proportioners, the final sample being collected in two air-tight sample containers of slightly over 1-quart capacity each. The balance of the coal, or reject portion, is deposited in a reject bin from which it is returned to the crusher feeder conveyors. There, the rejected portion mixes with the incoming coal and finally finds its way to the furnaces. One of the final sample containers is held at the plant as a reserve sample; the other is sent to the laboratory for moisture and Btu determinations and proximate analysis.

It has been found that some drying of the coal entering the plant occurs in the handling and processing between the weigh-point at the pits and the sampling point, the degree of drying depending upon the initial moisture content. For the most part such drying occurs in the crushing operation. In addition, there is also some further drying of the sample in the sample crusher mill and in the associated handling. Thus, the heating value determined from the laboratory sample is not precisely repre-

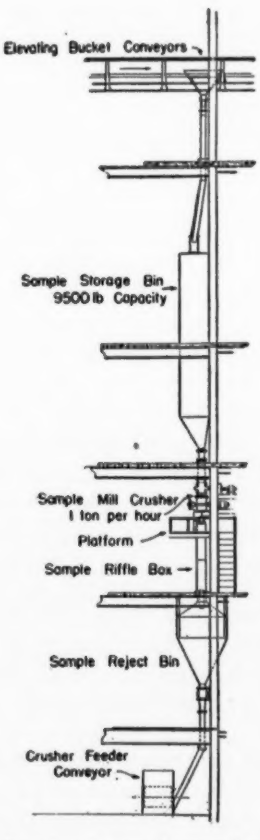


Fig. 28—Automatic coal-sampling system

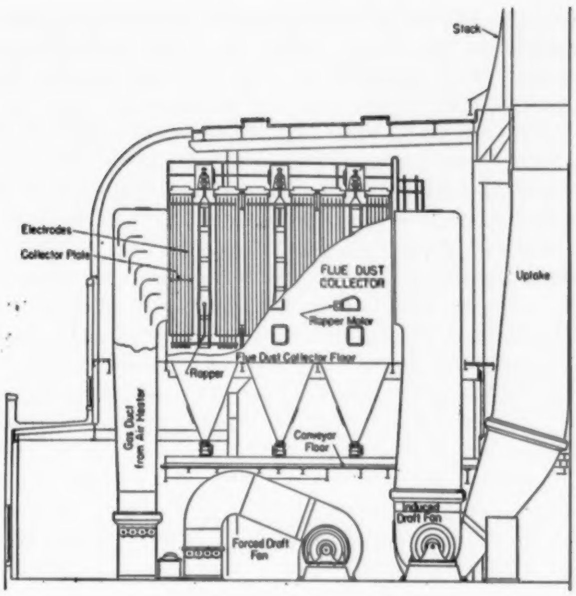


Fig. 29—Flue-dust collector

sentative of that of the coal being charged to the plant. Because of this, special samples of coal are taken monthly at the weigh-point and, therefore, before crushing and handling in the plant. From moisture values thus determined, a correction is established which is applied to the moisture content of the samples from the automatic sampler. This correction, on the average, is of the order of 0.5 per cent moisture.

Flue-Dust Collectors

The electrostatic flue-dust collector for each steam generator is located directly above the unit in an enclosure extending above the roof line of the original boiler house. The arrangement is such that the two collectors serving the two steam generators on opposite sides of the firing aisle will discharge into the same new stack. At present only the two steam generators in the new extension are equipped with collectors, delivery of the collector for the third new unit having been deferred because of the war. The installation of one of the collectors is shown in Fig. 29. Gases leaving the air heaters pass upward through the duct leading to the collector, then horizontally through the collector, then downward to the induced-draft fan and thence to the stack. Deflector vanes are installed in the elbows of the inlet and outlet ducts as an aid in effecting an even distribution of gas in the collector. As a further means of evenly distributing the gas, perforated plates with 2-in. holes on 3-in. centers completely cover the inlet and outlet openings of the collector. A photograph of the collector taken during erection and showing the perforated plate at the outlet end is shown in Fig. 30.

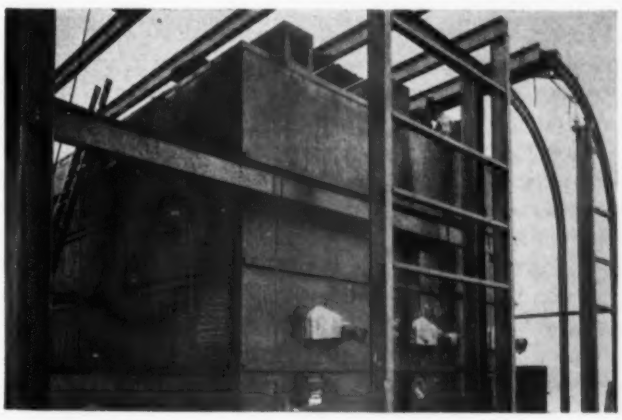


Fig. 30—Flue-dust collector during erection

Each collector consists of two sections installed side by side in the same housing; that is, looking at either end, the two halves on opposite sides of the vertical centerline are duplicates. Each section consists of three duplicate units, a separate ash hopper being provided for each unit as indicated in Fig. 29. There are, therefore, six ash hoppers for each collector.

The individual collector elements, or plates as they are called, are 3-ft wide and 17-ft 6 in. high. Each consists of two thin steel sheets (0.049 in. thick) supported face-to-face by a steel frame which holds them $^1/_2$ in. apart. These sheets are perforated over their entire area with $^1/_4$-in. holes on $^3/_8$-in. centers. The plates are installed on $8^3/_4$-in. centers in rows across the width of the housing, the surfaces of the plates being parallel to one another and to the direction of gas flow. There are two such rows to each of the three units of each section. Across the width of the collector the plates form thirty-four ducts for the passage of gas. The length of these ducts, that is, the length of the collector from inlet to outlet, is about 28 ft. Gas velocity through the collector at the rated maximum load on the steam generator of 440,000 lb of steam per hour is of the order of 500 fpm. Down the center of each duct are the discharge electrodes which consist of 0.109-in. diameter iron wires strung vertically and electrically connected to the high-voltage source. Between adjacent plates in each row there are five such wires on 5-in. centers.

Fig. 31—End view of furnace-bottom ash hopper

The distance between the adjacent edges of the two rows of plates in each unit of each section is made sufficient to accommodate a rapping device. Each of these devices consists of a horizontal shaft supporting pendulum-like weights or hammers suspended by semi-rigid arms. The shaft can be oscillated by operation of a motor-driven mechanism which causes the hammers to knock against steel cross members that tie adjacent

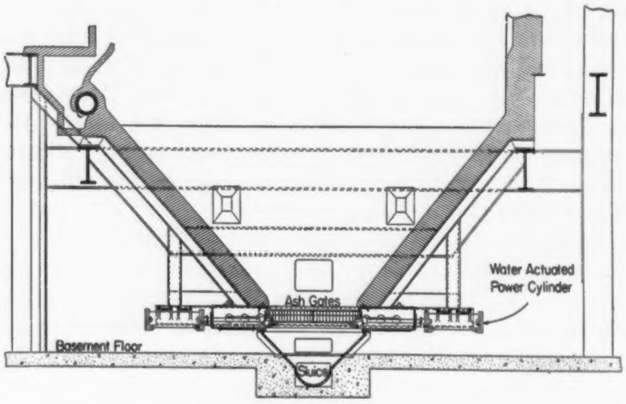


Fig. 32—Furnace-bottom ash hopper and gates

plates together in pairs at their lower ends. The vibration thus set up in the plates causes adhering ash to fall into the hoppers below.

The electrical equipment associated with each collector includes two three-phase, 60-cycle transformers, and two half-wave mechanical rectifiers driven by synchronous induction motors for converting the high-voltage secondary current of the transformers to undirectional current. The transformer primary is rated at 230 volts and primary control equipment makes it possible to obtain a range of secondary voltage up to 75,000 volts. This voltage, after rectification, is applied between the wire discharge electrodes and the plates. The wires are made negative and the plates positive. Ash particles in the gases pick up a negative charge from the discharge electrodes and are therefore attracted to the positively charged plates to which they adhere.

Ash Handling

Although by far the largest portion of the ash in the coal burned in the new Marysville steam generators finds its way to the flue dust collectors, some is collected at the bottom of the economizer and boiler passes, and another portion is collected at the bottom of the furnace. Each furnace has two hoppers side by side across the width of the unit. An end view of one of the hoppers is shown in Fig. 31. Ash which is collected in these hoppers in the dry state is discharged through gates to a water-sluicing system which transports it to a settling pit or sump. The ash gates are of the same general design as those used at the Trenton Channel Plant and are moved horizontally by water-actuated power cylinders. A cross-section of a hopper showing the gate arrangement is given in Fig. 32.

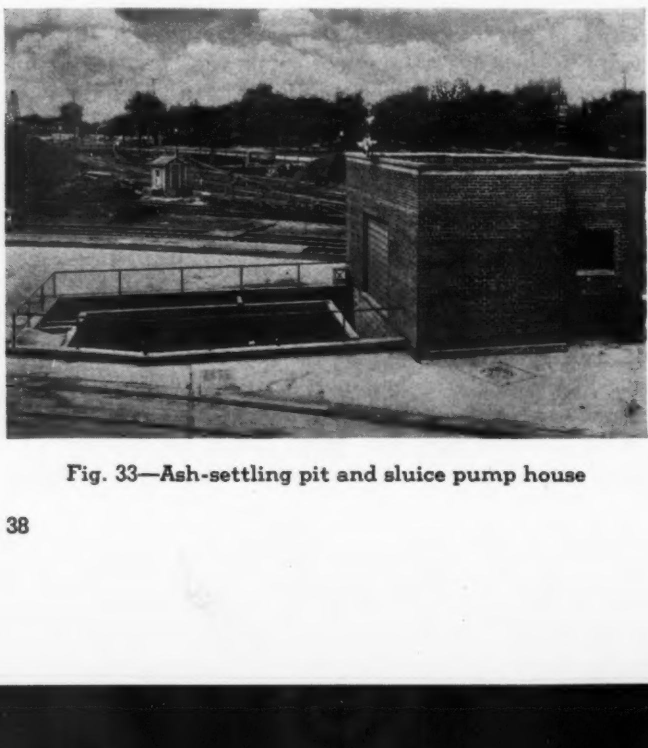


Fig. 33—Ash-settling pit and sluice pump house

A separate sluice trough is provided for each of the two rows of steam generators. The trough is semi-circular in cross-sections, 24 in. across the top and is formed of alloy-iron plates. Sluicing action is obtained by high-velocity water jets spaced at intervals of about 40 ft along the straight runs and also at locations where the sluice trough changes direction. At each jetting point the bottom of the trough is stepped down 3 in. in the direction of flow and the jet nozzle is installed in the step riser with its nozzle directed parallel to the axis of the trough. The discharge rate of the jets along the straight runs is 125 gpm and that of the jets at the turns is 175 gpm. Water is supplied by either of two pumps, each having a rated capacity of 1900 gpm and a discharge pressure of 140 psi. At present, the water requirement is such that the pump is run at a rate of 1575 gpm and about 115 psi at the pump discharge. The design capacity of the sluice in terms of ash is $^3/_4$ ton per minute.

The sluice system discharges into a settling pit located outside the plant from which the overflowing water is

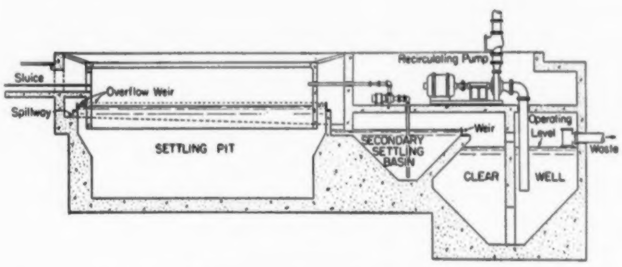


Fig. 34—Cross-section of ash-settling pit

recirculated to the sluice. A view of the pit and the sluice pump house is shown in Fig. 33. Associated with the pit is a secondary settling basin and clear water well as shown in Fig. 34. Most of the ash upon entering the pit settles quickly to the bottom. The overflowing water is made to pass beneath a suspended baffle wall into a trough running all around the pit. The purpose of the baffle is to avoid carrying along the comparatively fine ash particles that tend to float on the surface of the water. From the trough, the water passes over a notched weir into a spillway adjacent to the trough. The water flows from the spillway into the secondary settling basin which affords further opportunity for any entrained ash to settle out. From this basin the water flows over another notched weir into the clear water well from which it is lifted by the recirculating pump. Ash that settles in the secondary settling basin is pumped back into the main pit by a small pump as indicated in Fig. 34.

Periodically the ash is removed from the pit by a bucket crane and loaded into a rail car for transport to a nearby fill. Means are provided for draining the pit when desired, but for the routine removal of ash the water is not drained off. The capacity of the pit, secondary basin and clear water well at operating levels is 55,000 gallons. Normal makeup water to maintain the operating level is provided from the house-service system through float-operated valves.

In order to avoid scoring of the recirculating pump

shaft sleeves by any ash remaining in the water drawn from the clear well, two small high-pressure pumps are provided which inject relatively clean house service water into the pump shaft gland seals, thus preventing leakage of sediment-carrying water into these parts. Other pumps installed for the purpose of handling ash-laden water are similarly protected by water piped directly from the house-service system. Each pump is provided with a pressure-operated switch to open the motor circuit in case the sealing water pressure drops below the pump discharge pressure.

An isometric view of the system for transporting the fly ash from the flue-dust collector hoppers is shown in Fig. 35. Ash in the hoppers is discharged by gravity through rotary shut-off valves into a closed circuit collecting conveyor of the drag-link type. The arrangement is shown in Fig. 36. The conveyor deposits the ash in the hopper of the pneumatic transport pump shown in Fig. 37. Compressed air for this pump is furnished by the compressor shown in Fig. 38. The pump and compressor have sufficient capacity to handle the ash from four flue-dust collectors. When more than four new steam generators and flue-dust collectors have been installed in the plant, the transport pumping facilities will be duplicated. Ash from the flue-dust transport pump is carried away by a 5-in. pipe line.

For several years, the Company has studied the development of uses for fly ash in connection with the disposal of this material at the Trenton Channel Plant. A number of uses have been developed, some of which have provided outlets for sizable quantities. For the most part these uses require ash in the dry state. Accordingly, at Marysville a dry flue dust storage bin of some 1500-ton capacity has been provided alongside the boiler house. This is the circular structure shown in Fig. 39. The attached structure houses the associated flue-dust handling equipment. An isometric view of these facilities is shown in Fig. 40.

The 5-in. ash-transport line leading from the ash-transport pump in the boiler house is connected to a two-way valve at the top of the structure by means of which the ash can be diverted either into the storage bin or to the loading hopper. Ash to be hauled away from the plant in the dry state is loaded into suitable containers or closed conveyances by flexible hose connected to a pipe

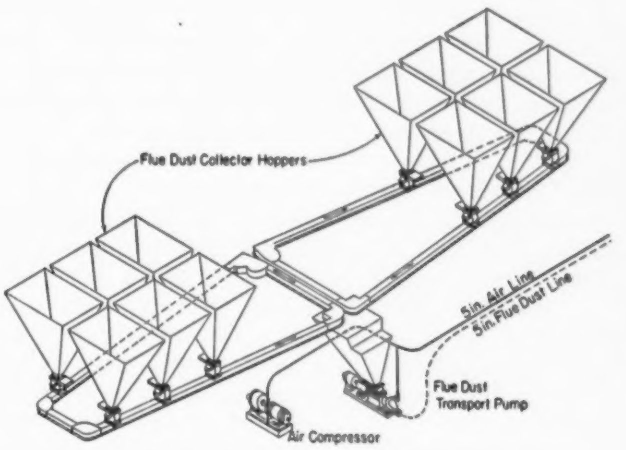


Fig. 35—Arrangement for transporting flue dust from collectors

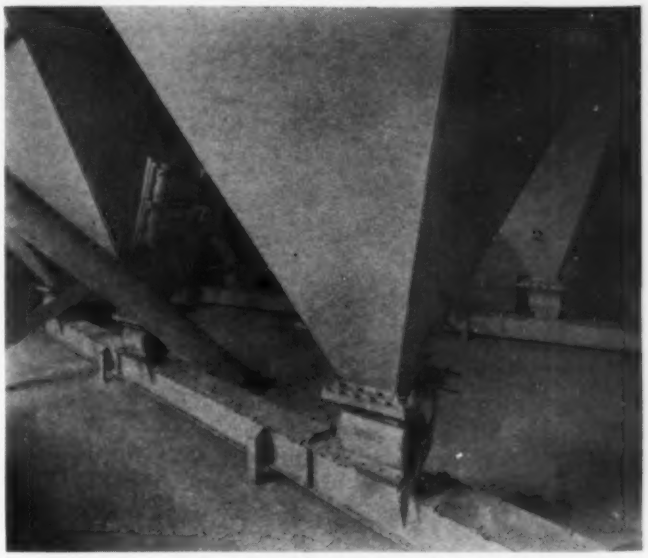


Fig. 36—Flue-dust collector hoppers and collecting conveyor

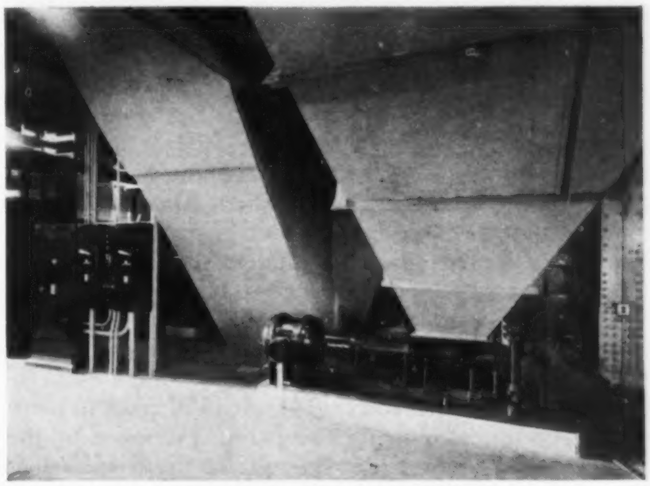


Fig. 37—Flue-dust transport pump

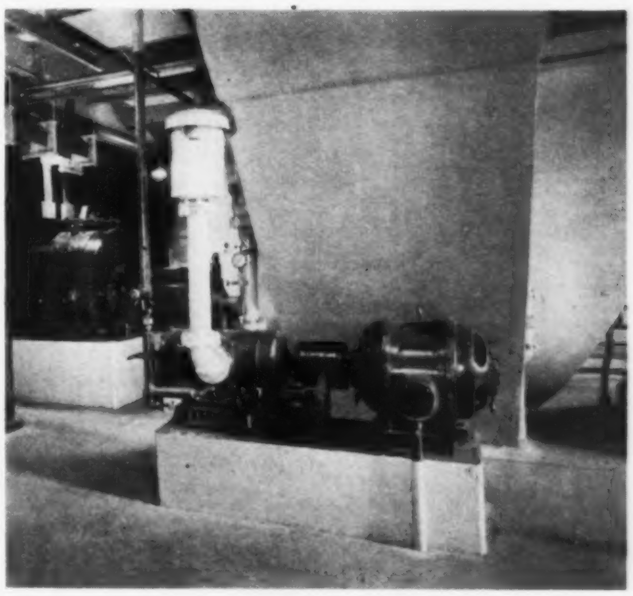


Fig. 38—Air compressor for flue-dust transport pump

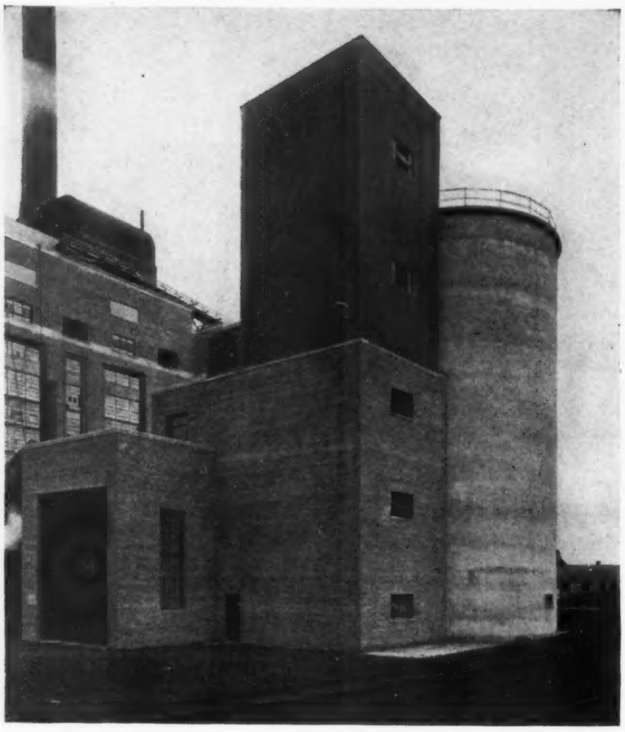


Fig. 39—Flue-dust storage bin and associated ash-handling house

leading from the loading hopper. The ash flows from the hopper by gravity. To draw ash from the storage bin, the material is fed into a pneumatic transport pump located at the bottom of the bin, as shown in Fig. 40, which transports the ash through a 5-in. pipe line up to the loading hopper previously referred to. This pneumatic transport pump is mounted on a narrow-gage rail truck so that in the event additional storage bins are built in the future, the same pump can be used to serve them. Compressed air for this pump is supplied by the same compressor that furnishes air for the transport of the ash from the flue-dust collectors. Air connection to the pump is by means of a flexible hose.

When there is no demand for dry ash, the material from the loading hopper is passed through special dampening devices, called dustless unloaders, so that it can be hauled away in open-type rail cars or trucks and used for fill. There are two such dampeners as indicated in Fig. 40. A photograph of the equipment, which is installed directly below the loading hopper, is shown in Fig. 41. The middle hopper spout seen in the photograph is the one used for loading dry ash into special containers or closed conveyances as previously mentioned. The dampeners are motor-driven devices in which ash from the loading hopper is tumbled in the presence of fine water sprays and then dumped into conveyances below. Sufficient water is added to prevent the ash from blowing away and causing a dust nuisance.

Turbine-Generator

The new Marysville turbine-generator is essentially a duplicate of the three machines installed in the high-pressure section of the Delray Power House. A photograph of the machine is reproduced in Fig. 42. It is a 75,000-kw, 1800-rpm unit and, as previously mentioned, is designed for 815-psi, 900-F steam at the throttle, and 1-in. of mercury back pressure. The unit weighs 1,125,000 lb and is mounted on a reinforced-concrete foundation which has no physical connection to any part of the building superstructure.

Turbine

The turbine has seventeen stages, the first stage wheel having a double row of buckets and the remaining wheels one row each. A sectional drawing is shown in Fig. 43. There are eight admission valves, four above and four below the horizontal centerline. The valve opening sequence is such that individual top and bottom valves open alternately, thus producing uniform heating of the unit.

The use of the eight control valves to admit steam to the different sections of the first-stage nozzle block, results in a comparatively flat economy curve over a wide range of load. The design heat rate of the unit, when

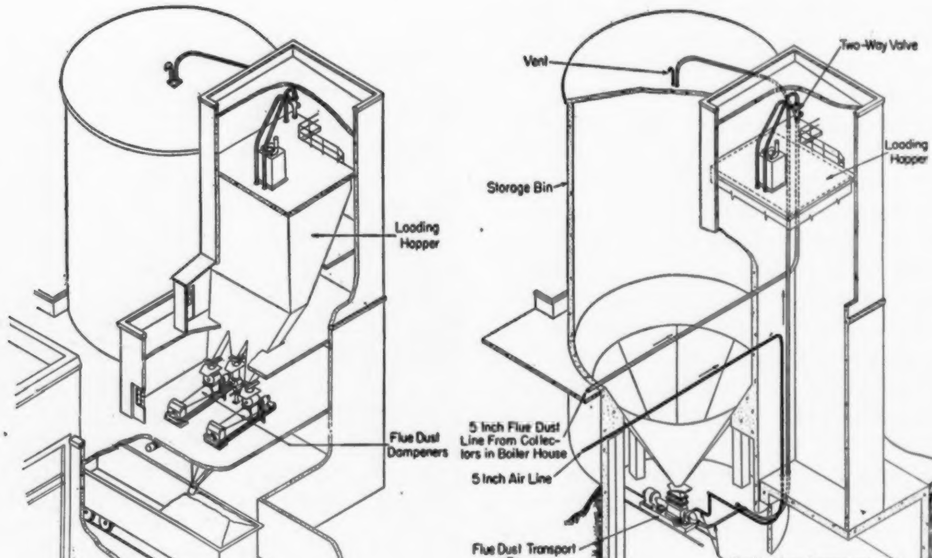


Fig. 40—Isometric view of flue-dust storage and handling facilities

operating at rated load conditions and with sufficient steam extraction to heat the feedwater to 390 F, is 9310 Btu per kwhr.

Particular attention was given to the general design and the materials used in the turbine oiling system in order to reduce the fire hazard to a minimum. The oil tank is located under the head-end pillow block. The main gear-driven oil pumps are located in the tank and are driven by a shaft extension from the front end of the turbine. The motor-driven auxiliary oil pump is mounted on the tank.

All oil piping to the front end of the turbine, valves, hydraulic cylinder, etc., is inside the head-end pillow block and oil tank, except the pipe to the stop valve which is enclosed in a guarding pipe. Thus, the high-pressure oil piping and joints are entirely enclosed and any leakage is effectively prevented from coming into contact with hot surfaces which might ignite the oil. Further, to avoid any hazard resulting from leakage from the low-pressure bearing-oil supply pipes, these are enclosed in the bearing-oil return pipes where the pressure is atmospheric.

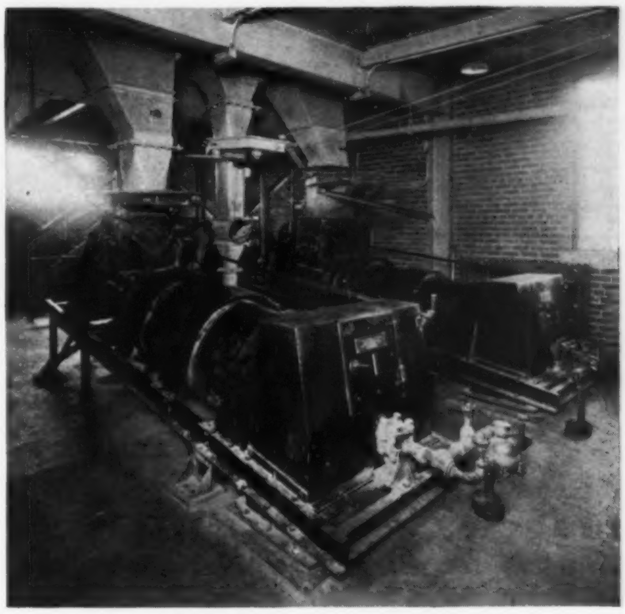


Fig. 41—Flue-dust dampeners

Generator

The generator is rated 100,000 kva, 0.75 power factor, 75,000 kw, 1800-rpm, four-pole, three-phase, 60-cycles, 14,400-volts, and has a short-circuit ratio of 1.0. It is designed to deliver full-rated kva at 0.75 power factor without exceeding temperature rise guarantees at any voltage between 5 per cent above and 5 per cent below normal. The temperature rise guarantees are 60 C on the armature and 85 C on the field. The insulation of the unit is, of course, Class B throughout.

The generator is air-cooled and, due to its size, four external fans are used for providing the necessary ventilation. The use of these fans results in a shorter and more efficient unit than could be obtained with fans directly connected to the rotor. The fans together with the air-cooler are located directly under the generator.

With only three ventilating fans in service, the generator can carry three-quarter load at rated power factor and can approach full kilowatt load if the power factor is brought close to unity. The power for these three fan motors is supplied by a transformer connected to the terminals of the generator; power for the fourth fan motor is supplied by the a-c house service system.

The air-cooling system for the main generator is also used for supplying the necessary ventilating air to the enclosed exciter and collector rings. However, any air which is circulated through the exciter and over the col-

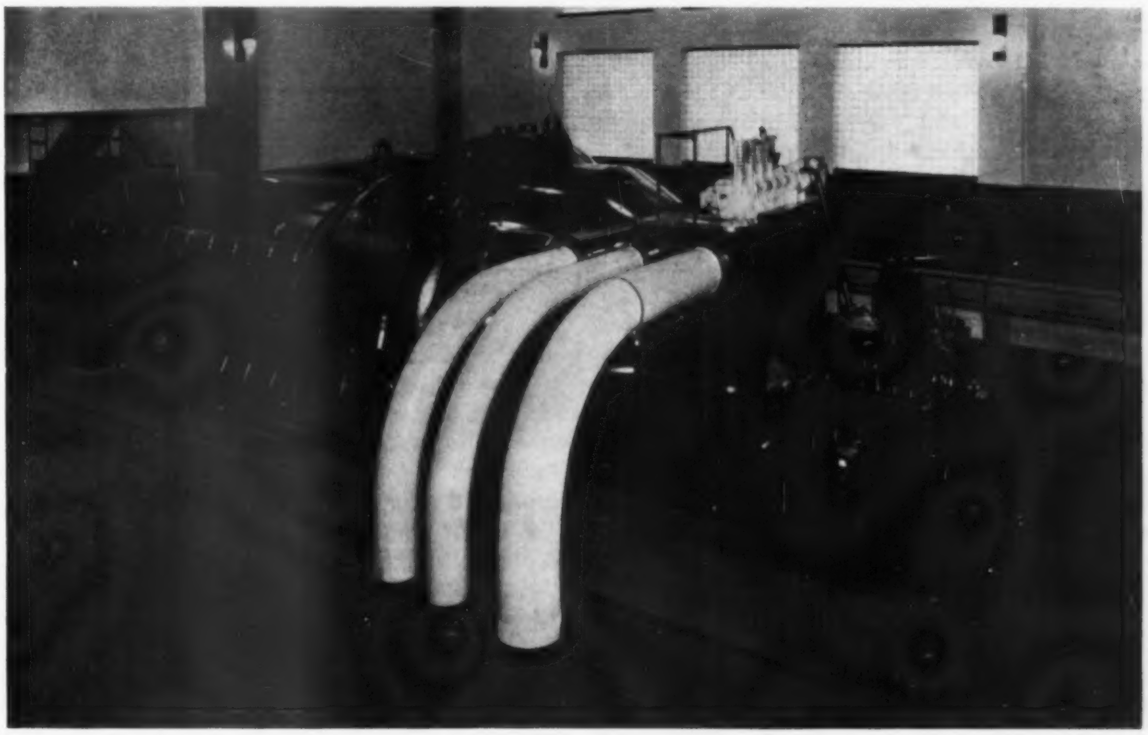


Fig. 42—New turbine-generator

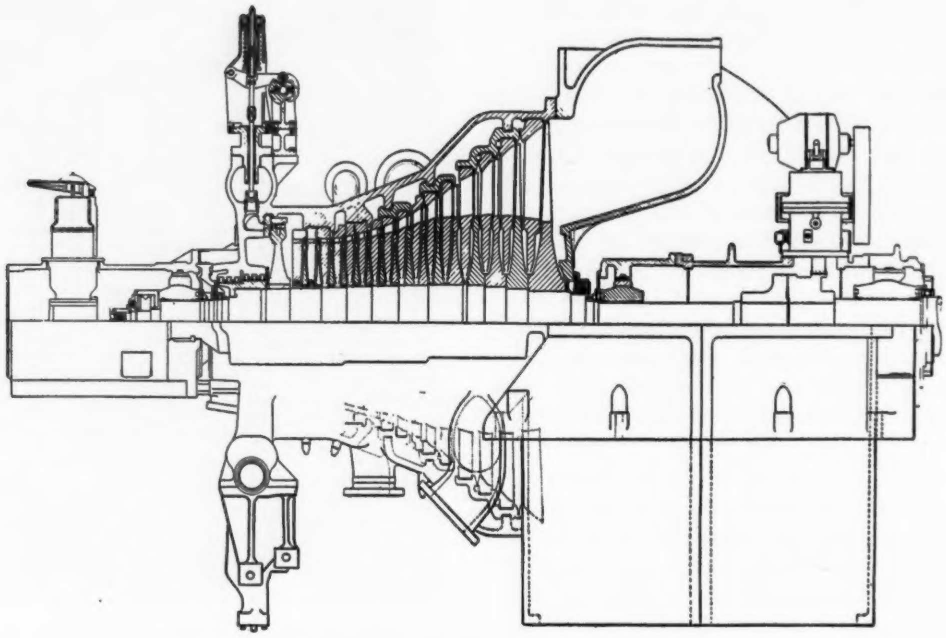


Fig. 43—Section through main turbine

lector rings is filtered before returning to the main generator circuit.

A direct-connected, overhung, 250-kw, 250-volt, d-c, self-excited stabilized exciter is used for supplying the field excitation. The exciter is stabilized down to about 65 volts. This voltage is below 90 per cent of the no-load, "cold" field voltage of the main generator. Thus, no difficulty should ever be encountered, even with hand control of the exciter, in synchronizing to the system under normal or emergency conditions.

Heat Balance and Auxiliary System Arrangements

The new high-pressure unit operates on a four-stage extraction cycle, without reheat, as shown in Fig. 44. This practice of bleeding the main unit was started experimentally in 1914 with the extraction of steam from a single-stage of each of two 9000-kw units at the original Delray Plant, and in later years has become common practice with the Company and throughout the central station industry.

The four extraction stages supply steam to four horizontal, closed-type, multi-pass heaters shown in Fig. 45. The tenth-stage bleed-point also supplies steam to a makeup evaporator, the vapor from which is discharged to the thirteenth-stage heater where it is condensed along with the thirteenth-stage extraction steam. Drains from the fourth and seventh-stage heaters cascade to the tenth-stage heater shell and thence in combination with the tenth-stage drain are removed by a heater-drains pump and returned directly to the boiler-feed circuit. Drains from the thirteenth-stage heater, including the condensed evaporator vapor, pass to the main condenser through a vertical, closed-type, multi-pass heat exchanger shown in Fig. 46, in which the drains give up heat to the condensate on its way from the main condenser to the thirteenth-stage heater.

The flow diagram, Fig. 44, shows that condensate from the condenser hotwell passes first to a low-head submerged-impeller, constant-speed condenser pump which delivers the condensate directly to the suction of a heater pump. The heater pump, operated in tandem with the boiler-feed pump by a common drive, delivers the condensate to the heat exchanger and thence through the thirteenth- and tenth-stage heaters to the suction of the boiler-feed pump. From the boiler-feed pump the water at boiler feed pressure is passed through the seventh- and fourth-stage heaters and enters the boiler-feed header.

Steam leaking off from the high-pressure shaft-seal packing and from the control valve-stem packings is piped to the heaters. The outer shaft-seal steam and outer valve-stem leak-off steam is carried to the thirteenth-stage heater. The seventh-stage heater receives the leakoff from the inner shaft seal, and the fourth-stage heater receives the steam from the inner valve-stem leakoff. Inasmuch as the thirteenth-stage pressure is always below atmospheric, a pressure gage and hand-operated throttling valve are provided in the leak-off line to the thirteenth-stage heater so that the outer shaft seal and outer valve-stem gland may be operated under a slight positive pressure. This pressure is regulated manually to about 1 psi g to prevent in-leakage of air.

Feedwater Heaters

As determined by the water pressure, the thirteenth- and tenth-stage heaters are classed as low-pressure heaters, while the seventh- and fourth-stage heaters are called high-pressure heaters. The statistics of operation of the heaters, together with the associated drains heat exchanger and the evaporator, are given in Table 1. The conditions refer to rated load on the turbine.

All four heaters and the heat exchanger incorporate floating-head construction. In the low-pressure heaters and drains heat exchanger, the floating-head cover is constructed of cast steel with integrally cast flange. In the high-pressure heaters, the floating head incorporates the manufacturer's "Lockhead" construction in which independent means are provided to care for the pressure load on the head and the joint load on the head-sealing gasket.

TABLE 1—FEEDWATER HEATING SYSTEM OPERATION, 75,000-KW DESIGN LOAD

	Stage Number					
	Drains Heat Exchanger	13th	Evaporator	10th	7th	4th
Steam						
Psi abs	...	7.3	34.8	34.8	100.6	220.0
Btu/lb	...	1120.4*	1182.3	1182.3	1284.2*	1334.4*
Lb/hr	...	46,810	12,060	40,560	46,300	47,700
Water						
Psi (rated)	250	250	...	250	1200	1200
F. inlet	79.3	89.0	50.0	173.7	258.6	328.2
F. outlet	89.0	173.7	...	255.9	328.2	392.8
Lb/hr	550,040	550,040	10,300	550,040	684,600	684,600
Passes	2	4	...	4	4	4
Surface						
Number of tubes	556	1344	...	1156	1568	1872
Square feet	477	2760	244	2370	3090	3575
Tube diameter OD, in.	5/8	5/8	1	5/8	5/8	5/8
Tube-wall gage	18 BWG	18 BWG	16 BWG	18 BWG	16 BWG	16 BWG
Tube material	Admiralty	Admiralty	Admiralty	Admiralty	Cupro-nickel	Cupro-nickel
Arrangement						
Vertical or horizontal	Vert.	Hor.	Hor.	Hor.	Hor.	Hor.
Desuperheat zone	...	No	No	No	Yes	Yes
Drip discharge to	Condenser	Drains heat exch.	13th	Heater drains pump	10th	7th
Terminal difference, deg F	...	5	...	3	0	−3

* Btu per pound entering heaters is higher than that of steam leaving turbine stages by reason of addition of gland steam and valve-stem leakage.

Tube bundles of the high-pressure heaters are fitted with baffles which isolate a part of the last water pass from the main volume of the steam space in the heater. The incoming bled steam is superheated and, since it enters the heater by way of this isolated section of the tube bundle, the water temperature is raised to a value equal to or exceeding the saturation temperature corresponding to the shell pressure. These conditions are indicated by the terminal difference values of 0 and −3 for the seventh- and fourth-stage heaters, respectively, in Table 1.

Evaporator

The evaporator is of the horizontal submerged-tube type with a capacity to produce 10,300 lb per hr of distilled makeup with a shell vapor pressure of about 11 psi absolute when the evaporator is supplied with 12,060 lb per hr of steam at the tenth-stage pressure of 34.8 psi absolute. It is guaranteed that the distilled water condensed from the evaporator vapor will contain not more than 5 ppm of solid impurities.

Heater Drains

The use of float-operated condensate drainers in the Company's older installations to control the flow of heater drains as they cascade from one heater to the next has been attended by operating difficulties with the drainers, and in some cases, severe erosion of the drain piping. An investigation of these difficulties was carried on within the Company[1] with the idea of substituting orifices in the drain lines for float-operated traps. In this investigation "it was found that erosion was a function of the amount of flashing and the quantity of water flowing in the pipe," and concerning the proposed use of orifices it was stated as an opinion that, "such an application would eliminate the operating troubles with traps and, if the orifices were installed at the end of the drain line rather than at the beginning, would prevent erosion in the piping." As a result of this study, orifices have been installed in place of several of the drainers

[1] "The Flow of Saturated Water Through Throttling Orifices" by M. W. Benjamin and J. G. Miller, *Transactions* A.S.M.E., July 1941, Vol. 63, No. 5, page 419.

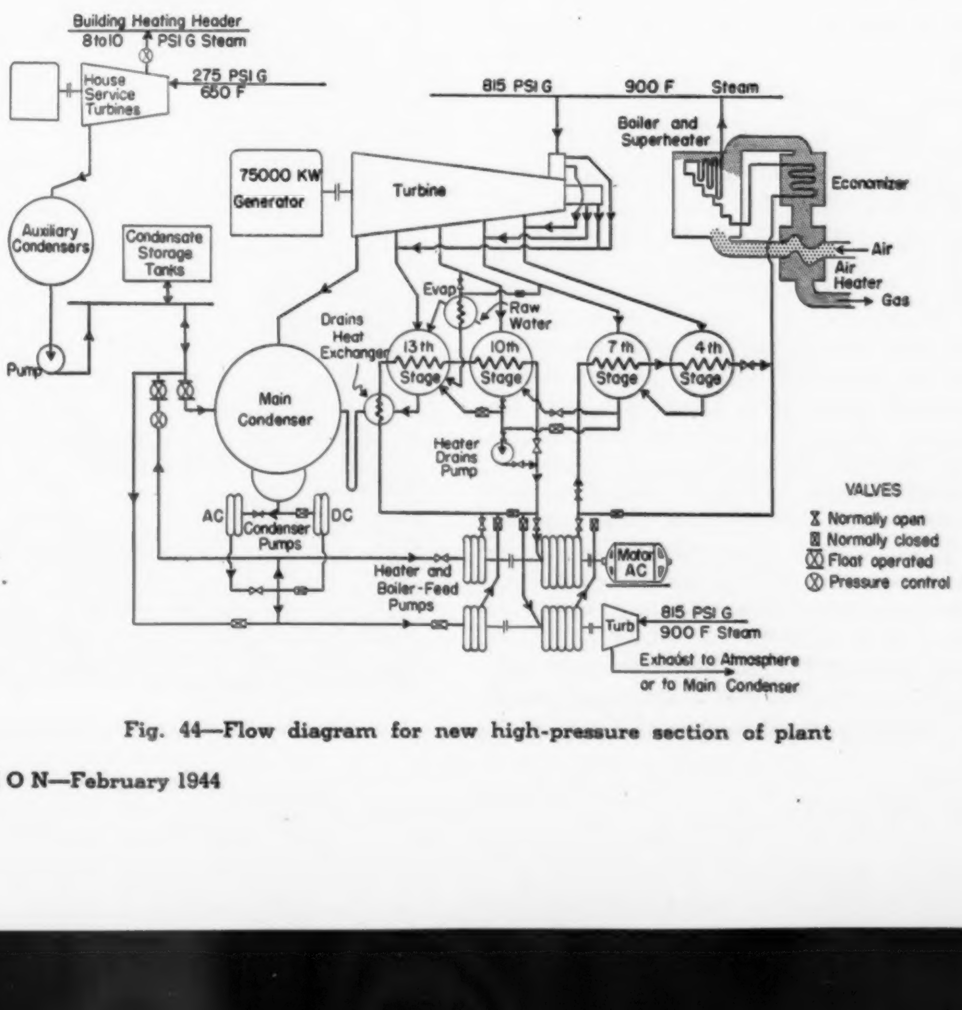


Fig. 44—Flow diagram for new high-pressure section of plant

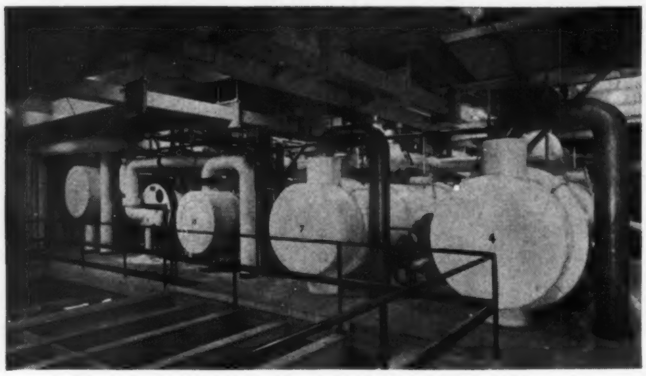


Fig. 45—Feedwater heaters and evaporator

which served the feedwater heaters of the Conners Creek units and of the first 75,000-kw machines at Delray. The success of this arrangement dictated the use of orifices from the outset with the second and third 75,000-kw units at Delray and also with the new one at Marysville.

Actually, with the heater arrangement employed, orifices are required only for the drains from the fourth- and seventh-stage heaters. The drains from the tenth-stage heater combined with those cascaded from the fourth- and seventh-stages are removed by the heater drains pump, which in itself acts as a flow controller.

The evaporator tube bundle drains to the thirteenth-stage heater through a float-operated trap. An orifice is not well suited to this application because the relationship between turbine load and evaporator output is indefinite, varying with the amount of scale adhering to the tubes. Furthermore the rate of condensation in the tube bundle is abnormally high during a scale-cracking cycle when, with the shell filled with cold water, steam for cracking scale is admitted from the seventh-stage bleed point.

All heaters have the shell, or steam space, vented to the main condenser for the purpose of removing non-condensable vapors which are carried in the steam from the bleed point.

Oxygen Control

Considerable attention was given to the problem of maintaining oxygen-free feedwater for the new Marysville steam generators. It is the practice of the Company to employ the main-turbine condensers as deaerators. Flashing and venting, which are necessary for oxygen elimination in any apparatus, can be handled efficiently in a surface condenser simply by admitting the water into the steam space through spray manifolds. Temperature of the incoming water preferably should exceed the saturation temperature corresponding to the condenser pressure. Flashing under vacuum in the condenser is followed by venting off the non-condensable vapors to the condenser air discharge. All water and steam which might become contaminated with oxygen or other gases are passed through the condenser for deaeration before being admitted to the boiler-feed circuit. For example, the condensate from the thirteenth-stage heater, including the condensed evaporator vapor, is returned to the feedwater system by way of the main condenser because the thirteenth-stage heater operates at subatmospheric pressures at all loads and therefore is subject to in-leakage of air. Condensate from the open storage tanks, which is another carrier of dissolved oxygen, is deaerated in the main condenser.

Miscellaneous condensate, amounting to about 5 per cent of the total boiler feed, also is deaerated before its return to the plant cycle. The largest part of this condensate comes from steam used by the house-service turbines, while the remainder includes shaft and valve-sealing water, building-heating condensate, various hot and cold drips, and the condensate from the hydraulic operators of the boiler-feed regulating valves.

The amount of oxygen in the discharge of the heater-drains pump is negligible, since the fourth-, seventh- and tenth-stage heaters all operate at pressures greater than atmospheric at all turbine loads above 35,000 kw. The pump is not used when the load is less than 35,000 kw, in which case all the heater drains are cascaded to the main condenser.

Plant Drips

The quantity of condensate represented by plant drips justifies a rather extensive system of piping to collect and return them to the boiler-feed circuit. In the low-pressure section at Marysville, which will continue to operate, hot drips are returned to the "hot" boiler-feedwater storage tanks. For the time being, and until additional high-pressure machines are installed, it was decided to continue using these "hot" tanks as the collecting center for hot drips returned from the high-pressure section also.

Cold drips from the main turbine and its auxiliary equipment are returned to a small collecting tank from which they are discharged to the main unit condenser for deaeration and return to the boiler-feed system.

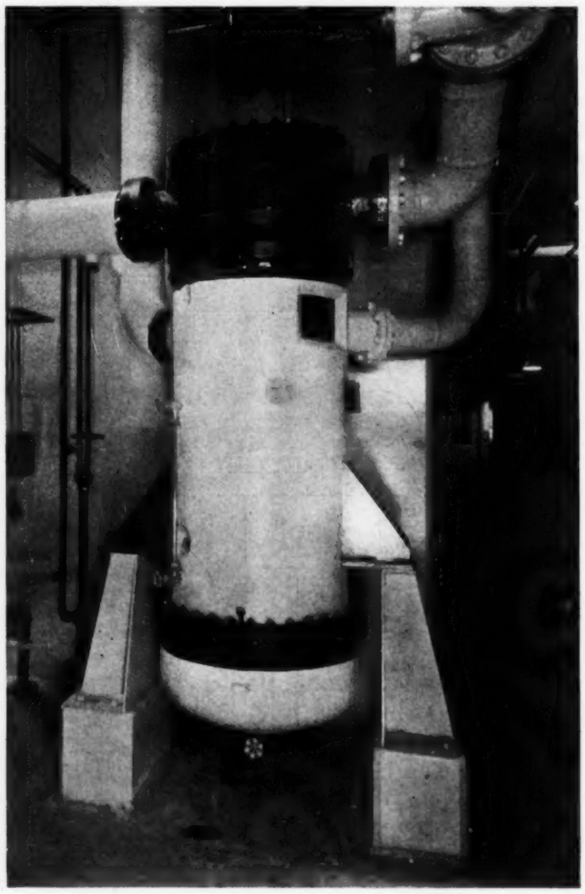


Fig. 46—Vertical drains heat exchanger

Condensate Storage

The amount of condensate held within the plant cycle varies with plant load and, in order to accommodate this variation, it is necessary to provide surge capacity somewhere in the condensate system. In addition to this requirement it is desirable to have a quantity of condensate in reserve to accommodate unusual demand for water, such as would occur in case of a boiler-tube failure.

In the old plant, because of the nature of the plant cycle, these needs were met by two separate sets of tanks, one for hot and one for cold condensate. With the installation of the new turbine-generator some modifications of the old storage system were necessary, and, as matters now stand, the condensate storage capacity for the plant as a whole amounts to 668,000 lb, or 2.97 lb per kw.

Introduction of Storage Condensate Into the Flow Circuit

The condensate and boiler feed-pump system takes its suction from a hotwell which is bolted directly to the bottom of the condenser shell as shown in Fig. 47, and has a normal water-level capacity of 41,000 lb. Two 6-in. float-operated valves, one of which is pictured in Fig. 48, maintain the normal water level. When the demand for water exceeds the rate of condensation in the main condenser, the water level in the hotwell falls, and one of the valves opens to admit water from the condensate-storage system to make up the deficit. This water is discharged into the steam space of the condenser through a system of four steel spray pipes. Two of these, one at either side of the condenser shell, run horizontally along the side at the elevation of the tube-head centerline. Each pipe is 3-in. ID and 20 ft long with capped ends and has $^3/_{16}$-in. spray holes spaced at $2^3/_4$-in. intervals along the top so directed that the water does not impinge on the condenser tubes. These pipes care for all normal introduction of water up to an input of 100,000 lb per hr. However, to provide for greater amounts up to a total rate of 400,000 lb per hr, two additional spray pipes, one at either side of the condenser shell, are installed 8 ft above the tube-head center. The upper pipes are connected to the same incoming 6-in. connection from the low-level float valve and act merely as an overflow or supplementary point of admission for makeup water. Each upper elevation pipe is 3-in. ID and 17 ft long, with capped ends, and has $^1/_4$-in. holes with 4-in. spacing. The water discharged through these holes is also directed upward in a manner that avoids impingement on the condenser tubes.

Fig. 47—Welded steel condenser hotwell

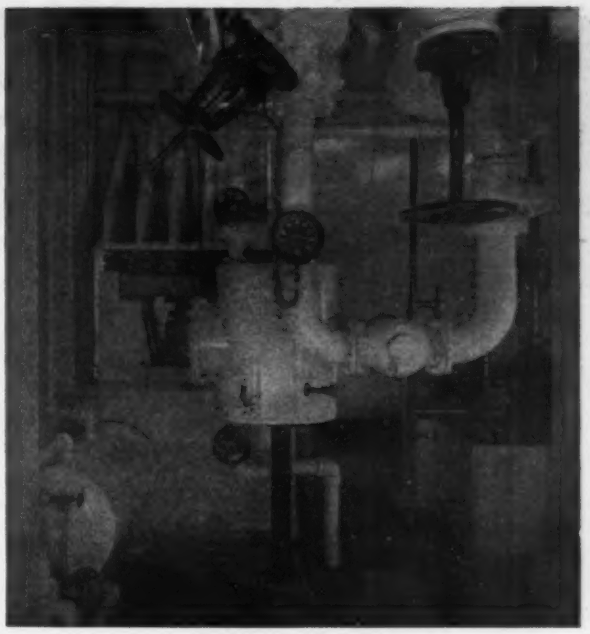


Fig. 48—Float-operated condensate makeup valve

Diversion of Condensate from Flow Circuit to Storage

Makeup water from the storage system enters the condenser in some amount almost continuously while the turbine is in operation. However, there are times when the throttle steam flow exceeds the boiler-feed flow and the hotwell level rises. Under such conditions, the low-level float valve closes and the high-level float valve opens and permits some portion of the total condensate handled by the condenser pump to be diverted from the flow circuit to storage. The flow diagram given in Fig. 44 shows a minimum-pressure regulating valve installed at the inlet side of the high-level float-operated valve. This device is set to maintain a pressure which will always exceed 25 psi at the heater-pump suction. The purpose of the valve is to prevent a reduction of pressure in the line to a value at which flashing could take place in the heater pump and possibly carry through to the boiler feed-pump suction. This condition might otherwise occur in case the high-level float valve were full open, because the combined static and friction head of discharge to storage is of the order of only 17 psi g.

*High-Pressure Steam for Marine Propulsion**—II*

By **R. H. Tingey, Chief Engineer, Fore River Yard, Bethlehem Steel Co.**

The first part of this exhaustive analysis of the thermal and economic gains through employment of high pressure and high temperature steam, compared with present standard practice for merchant vessels, appeared in the January issue. The length of the paper made it necessary to continue the text to the February issue in which various cycles are compared and an Appendix included, describing the method of calculating the turbine condition curves and performance of the actual cycles.

Fig. 10, derived from Fig. 8, shows the reduction of oil consumption due to the use of various cycles as compared to a plant with standard steam conditions, 425 psi 740 F. Note that with constant steam temperature the gain in fuel consumption for an actual installation has a maximum at some finite pressure but that there is no such maximum for the ideal steam cycle as shown on Fig. 4. This deviation of the actual efficiency from the ideal is due to the additional moisture loss, turbine parasitic losses, feed pump loss, etc., which increase at an accelerated pace as pressure is increased.

Losses due to pressure drop through the reheater and piping have been included in the reheat-cycle curves of Figs. 8 and 10. In some cases the reheat pressure is not exactly that which gives maximum efficiency but has been selected with view to the practical limitations imposed by an actual installation. For example, it is desirable to place the reheater between the exhaust of one turbine and the inlet of the next casing in the series to avoid the complication that would result if steam were withdrawn from the middle of a turbine casing and then returned to it reheated. As the receiver pressures must be selected to give proper division of work between the pinions of the reduction gear, this imposes some limitation on the choice of reheat pressure, but the loss incurred is in no case over 1 per cent. Another case arises with gas reheat where the selected reheater pressure is made somewhat higher than that for maximum efficiency to reduce the pipe size. This piping is a prominent feature of the installation that should not be minimized.

These curves show that the oil consumption is reduced about 4 per cent when the initial pressure is doubled, keeping the initial temperature constant. This approximation is correct unless the exhaust moisture becomes excessive, i.e. over 13 per cent. Likewise a 4 per cent improvement in economy is made when the initial temperature is increased 100 deg F, keeping the pressure constant. These gains due to pressure and temperature are additive in most cases.

When reheat is used, increasing the initial temperature 100 deg F reduces the oil rate 2 per cent. Likewise, increasing the reheat temperature 100 deg F makes a 2 per cent improvement. These gains are additive so that if both initial and reheat temperatures are increased 100 deg F a 4 per cent gain is made. These gains are additive to the pressure gain of the previous paragraph also.

The addition of one stage of gas reheat to a high-pressure cycle has roughly the same effect on economy as increasing the initial temperature 200 deg F.

Marine engineers cannot always agree as to the efficiency attainable with given steam conditions but there is not much room for difference of opinion as to the gains obtainable by varying the steam conditions and therefore Fig. 10 probably forms a better basis of discussion than Fig. 8.

Discussion of Practicable Cycles

Many of the cycles indicated on Fig. 10 are impracticable due to excessive moisture at the exhaust end or at some other point within the turbine. These have been eliminated and Table 2 has been compiled listing the more important remaining cycles along with the gain in oil consumption obtained by each. Only initial pressures that utilize existing ASA standards to the limit allowed by the Coast Guard Rules have been listed and three initial temperatures 740 F, 840 F and 940 F have been selected for discussion. These steam conditions amply cover the useful range. The best of these cycles have been selected by the process of elimination and are listed in Table 3.

Turbine condition curves on the enthalpy-entropy diagram for some of these cycles are shown on Fig. 11. These graphically illustrate some of the principal features of the cycles; for example, the relative amount of work done in the superheated and moist regions which determines the moisture loss. Their principal use is to show the variation of temperature and pressure through the turbines and reheaters. Note that a 2500-psi 740-F cycle is not practicable even with reheat because steam in expanding through the turbine would become moist

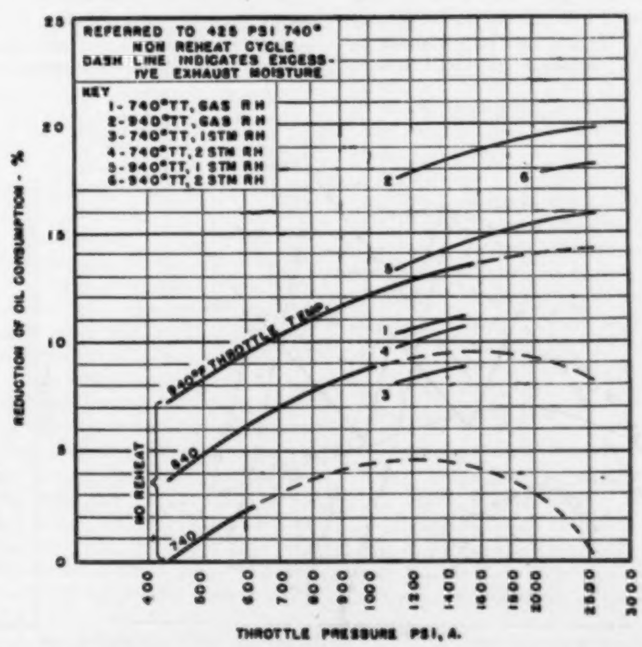


Fig. 10—Reduction of oil consumption for actual installations

* A paper delivered before the New England Association of Naval Architects and Marine Engineers, Oct. 26, 1943.

TABLE 2

Cycle No.	Throttle Pressure, Psi G	Throttle Temp., F	No. and Type of Reheaters	Reheat Pressure, Psi G	Reheat Temp., F	Red. of Fuel Rate, %
1	425	740	......	...	...	0
2	600	840	......	...	...	6
3	1000	840	......	...	...	9
4	855	940	......	...	...	11
5	1425	940	......	...	...	13
6	1425	740	1—Steam	200	570	9
7	1425	740	2—Steam	200, 30	570	11
8	1660	840	1—Steam	175	590	11
9	1425	940	1—Steam	30	570	14
10	1425	740	1—Gas	300	740	11
11	1660	840	1—Gas	300	840	15
12	1425	940	1—Gas	300	940	19
13	2780	840	1—Steam	300	660	..
14	2780	840	2—Steam	300, 40	660	..
15	2370	940	1—Steam	300	640	15
16	2370	940	2—Steam	300, 40	640	18
17	2780	840	1—Gas	300	840	..
18	2370	940	1—Gas	300	940	20

at too high a pressure for reheat to be used effectively.

Cycles 1 and 2 are of the conventional type utilizing superheat only but no reheat. They represent present practice and are included mainly for comparison with the others. Cycle 2 is considered a good way of obtaining the moderate gain of 6 per cent. However, this does not represent the best that can be obtained by using higher pressures. An initial pressure of 600 psi is suitable for lower powers as indicated on Fig. 1, and Cycle 2 is probably the best way of utilizing it in this case. A higher temperature might be used with 600 psi to give more gain if desired but reheating is not very beneficial at this pressure and would not be recommended.

Cycles 3 and 4 give a good gain in efficiency but are suitable only for somewhat higher powers as shown on Fig. 1. These 900-psi cycles are preferable to 1425 psi in the range of powers from about 6000 to 10,000 shp but for higher powers Cycle 7 is considered better because it gives the same gain in efficiency without the use of high temperatures. The use of reheat at this pressure would be a border line case and probably would not be justified.

Cycle 5 is of the conventional type also, but it incorporates higher than normal steam conditions and this gives a substantial gain in efficiency. However, almost as large a gain can be obtained by using Cycle 7, without the risks accompanying the use of high temperature. The boiler for Cycle 7 is of the conventional type with an integral superheater and no complications in design or operation. On the other hand, Cycle 5 requires some kind of superheat control to prevent the steam temperature from rising excessively during maneuvering and this complicates the boiler considerably. The simplicity of the boiler for Cycle 7 as compared to Cycle 5 compensates to a large extent for the complication introduced by the two steam reheaters. This fact, combined with the complete absence of all temperature problems, makes Cycle 7 more attractive than Cycle 5. However, Cycle 5 might be adopted by an owner who wants the highest efficiency practicable at the present time and is willing to take the risks that go with it.

Cycle 6 has been eliminated because the addition of one low-pressure reheater, which converts it into Cycle 7, is justified by the 2 per cent gain in economy obtained.

Cycle 7 is considered one of the best arrangements and will be discussed below.

Cycle 8 might be considered acceptable if the boiler can be of the conventional type with an integral superheater and no superheat control. This is possible if high temperature materials are utilized throughout the superheater, steam piping and high-pressure turbine. These parts must be designed to take the surges in temperature that occur when the vessel is getting underway and maneuvering, as under these conditions the temperature may approach 925 F for short periods. Cycle 8 may be given consideration when high-temperature materials are commercially available again, but Cycle 7 appears to be preferable because no temperature problems exist and the gain in economy is the same.

The machinery for Cycle 9 is the same as Cycle 5, except that a single stage of steam reheating has been added. The reduction of oil consumption due to this reheater (about $1^1/_2$ per cent) is too small to justify its use and therefore Cycle 5 is preferred to Cycle 9.

The oil consumption of a power plant using Cycle 10 is the same as with Cycle 7 In view of this, and also giving due consideration to the increased complication in the design, construction and operation of the gas reheat boiler, Cycle 10 is considered to be less desirable than Cycle 7.

A more complete discussion of gas reheat (Cycle 10) and a description of the machinery for the S.S. "Examiner," which operates on this cycle will be found in Vol. 49 of the *Transactions* of the Society of Naval Architects and Marine Engineers, 1941, in a paper entitled "A 1200 Pound Reheat Marine Installation." A report of service experience was published in the *Transactions* the following year.

Cycle 11 is 2 per cent more efficient than Cycle 5, but it is not so desirable from other points of view. The high temperature gas-reheat boiler for Cycle 11 is more complicated to build and operate than that for Cycle 5. Also high-temperature materials and constructions are more extensively used in Cycle 11, because both the superheated steam and the reheated steam are at high temperature. However, this is partly offset by the lower temperature used in Cycle 11. All things considered, the 2 per cent gain does not seem to be justified by the disadvantages incurred and therefore Cycle 5 is preferred to Cycle 11.

The efficiency of Cycle 12 is the best in the 1500-psi group and is exceeded only slightly by the best of the 2500-psi cycles. The gain in going to 2500 psi is not large enough to warrant adopting such a high

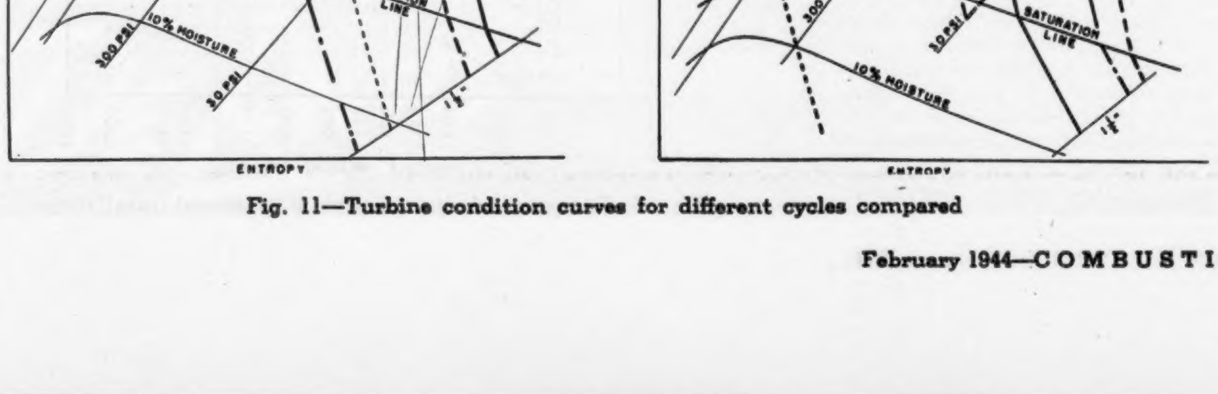


Fig. 11—Turbine condition curves for different cycles compared

pressure. Therefore, Cycle 12 sets a level of efficiency for a marine steam power plant that is not likely to be materially exceeded for a long time. The difficulties to be overcome in the application of Cycle 12 are severe because the more serious complications of all the other cycles are combined in this one. Both superheat and reheat control of some kind are necessary in view of the high temperature, 940 F, and some means of preventing the reheater from burning out when running astern must be provided also. It will be noted that there is much more equipment subject to high temperature with this scheme than with the others. This requires more heat-resisting material in the superheater, reheater, main steam piping, reheater piping and turbines, and greater care must be exercised in the design of these parts.

Cycle 12 is considered to be the most practicable scheme to obtain a very large increase of efficiency, i.e., 19 per cent over 425 psi 740 F. All of the important elements of this cycle except high temperature, have been utilized on the S.S. "Examiner." The machinery of this vessel operates on Cycle 10 which is similar to Cycle 12 except that it uses a lower temperature. The experience on this vessel to date is very encouraging and there appear to be no major unsolved problems pertaining to either high pressure or gas reheat. However, in adopting Cycle 12 the additional risks associated with high temperature must be taken.

The above discussion leads to the selection of the cycles listed in Table 3 as the most promising:

could later be combined with gas reheat to form Cycle 12 and so obtain another large increase of efficiency.

Cycle 12 is the most radical of those selected and is not recommended yet for general commercial service. This cycle should be considered experimental in view of the large risks involved and therefore it should not be adopted for a whole class of ships. However, Cycle 12 gives the highest gain in economy of the selected group and it can be built utilizing present knowledge, materials and practices. It is hoped that it will be incorporated in an experimental ship as soon as conditions permit such developments.

It should be noted that more than half of the maximum gain attainable with high steam conditions can be obtained by the use of Cycle 7 without appreciable risk. If a further reduction of oil consumption is desired the additional risks that must be taken are all out of proportion to the gains obtained. For example, Cycle 7 gives 11 per cent gain with negligible risk while Cycle 5 gives an additional gain of only 2 per cent but entails a considerably larger risk due to the use of high temperature.

Weight, Cost and Space

The dry weight of machinery for a 1425-psi 740-F installation with gas reheat is about 5 per cent more than standard machinery. Although no detailed estimates have been made of the weight with 1425 psi 740 F and two stages of steam reheat it will be about the same as standard machinery. Using 1425-psi 940-F machinery

TABLE 3

Cycle No.	Throttle Pressure, Psi G	Throttle Temp., F	Type of Reheater	No. of Reheaters	Reheat Temp., F	Red. of Fuel Rate, %
7	1425	740	Steam	2	570	11
5	1425	940	.....	0	...	13
12	1425	940	Gas	1	940	19

The final decision as to which cycle will be used in a particular ship will be influenced largely by the conservatism of the owner. Cycle 7 gives a respectable gain in economy without departing radically from proved practice and therefore is the most conservative choice. The important elements in this cycle have been tried out on the S.S. "Examiner" and the lessons derived from this experimental high-pressure installation can be incorporated in new construction now. Cycle 7 is ready for immediate adoption by a ship-owner who is willing to take very moderate risks for an 11 per cent gain. Bethlehem Steel Company, Shipbuilding Division, has a class of ships under construction using this cycle.

Cycle 5 is less conservative than Cycle 7 because it involves high temperature but the additional risk is not prohibitive. As soon as the alloying elements necessary to produce heat-resisting steels are again available for commercial vessels this cycle undoubtedly will be adopted by some owner who is looking for the best efficiency practicable at the present time. However, the gain in fuel rate as compared to Cycle 7 is scarcely worth the additional risk. The principal value in building a ship with Cycle 5 would be to obtain experience with high temperature which

without reheat the weight probably will be about the same also. It is expected that the machinery weight for the various other practicable cycles utilizing high pressure will be within the range of 0 to 5 per cent above normal. These weights are for a 10,000-shp installation. With larger powers the weight of the high-pressure machinery will approach more closely that of a normal-pressure installation and actually it may be a little less, because the size of some of the high-pressure equipment will not increase in proportion to the power.

Cost of high-pressure plants utilizing 740 F may be assumed proportional to the weight because the materials and the workmanship required are the same as for normal pressures. Cost of a high-pressure high-temperature installation will be somewhat greater than indicated by the weight because some of the material required consists of heat-resisting alloys which are more expensive than the carbon steel used with 740 F and also because some of the constructions required are more costly. Therefore, it is expected that an installation using Cycle 7 will cost about the same as a normal-pressure plant and that Cycles 5, 9 and 10 will cost about 5 per cent more. Cycle 12, which incorporates high pressure, high temperature and gas reheat, will undoubtedly cost more than this because there is a great deal more additional high cost material due to the additional equipment subject to high temperature. The development costs would be abnormally large for such an experimental installation but these would not apply to ships subsequently built with this cycle. In larger powers the differential in cost between high and normal pressures would decrease.

The adoption of 2370 psi would require the use of special constructions for some parts and the processes for their manufacture may not be in common use now. For example, the piping thicknesses required are greater than are now commercially available and this might require that they be manufactured from hollow-bored forging. This would increase the cost all out of proportion to the increased weight.

In a merchant ship ample space is usually assigned to accommodate the machinery due to the application of the tonnage rules. These rules were apparently written to insure enough space for working the engines when they were much more bulky than at present. The rules permit a 32 per cent deduction from the gross tonnage in obtaining the net tonnage if the machinery space exceeds 13 per cent of the gross tonnage. As harbor dues, canal dues, fees, etc., are based upon net tonnage it is usually to the interests of the owner to take advantage of this deduction by providing a machinery compartment a little more than 13 per cent of the gross tonnage, whether the machinery actually requires this space or not. All of the machinery installations discussed in this paper can be accommodated without crowding in a machinery space of this size. In the S.S. "Examiner" a good machinery arrangement was obtained in exactly the same space as that occupied by the normal-pressure machinery in her sister ships. A 1425-psi 740-F steam-reheat installation has been worked out in detail for a ship with machinery aft and there was no difficulty in obtaining a satisfactory arrangement.

If gas reheat is adopted it is essential that the boilers and turbines be fairly close together; otherwise the amount of reheater piping becomes excessive. Placing the turbines and boilers in the same machinery space is ideal for this type of machinery. The use of steam reheaters permits somewhat greater freedom in the relative location of boilers and turbines. However, even in this case it is advantageous to keep the boilers and turbines fairly close together because it reduces the length of steam supply and drain piping associated with the reheaters. With high pressure and high temperature no unusual limitations are imposed on the location of machinery.

Conclusions

1. A marine power plant using 1425-psi 740-F initial steam conditions and two steam-reheat stages is now beyond the experimental stage and is thoroughly practicable for powers of the order of 10,000 shp per shaft. It will have an oil rate of about 0.51 lb per shp per hr which is 11 per cent better than with standard steam conditions.

2. The oil rate of a plant with 1425-psi 940-F initial steam conditions and no reheat stage will be 0.50 lb per shp per hr which is 13 per cent better than standard. This power plant is feasible now but entails more risk due to high temperature.

3. An installation using 1425-psi 940-F initial conditions with gas reheat to 940 F has an oil rate of 0.46 lb per shp per hr which is 19 per cent better than standard. This scheme is suitable for an experimental installation aboard ship but not for general use yet.

4. It appears than an initial pressure of 2370 psi will not give sufficient gain over 1425 psi in the powers usually installed aboard merchant ships to justify its use. However, there has been insufficient development of equipment for 2370 psi to form the basis of a firm opinion.

5. For powers below 10,000 shp lower initial steam conditions should be adopted for the best overall economy as shown in Fig. 1.

6. The cost and weight of the 1425-psi machinery will probably be about 0 to 5 per cent greater than for 425 psi 740 F depending on the cycle adopted.

TABLE 4

Point	Enthalpy H	Enthalpy Drop ΔH	Turbine Flow Q	$Q\Delta H$
H.P. Inlet	1325			
H.P. Exhaust	1187	138	0.871	120
I.P. Inlet	1307			
I.P. Bleeder	1254	53	0.806	43
I.P. Exhaust	1174	80	0.734	58
L.P. Inlet	1319			
L.P. Bleeder	1189	130	0.734	95
L.P. Exhaust	1067	122	0.690	84
Summation		$\Sigma\Delta H = 523$		$\Sigma Q\Delta H = 400$
	Enthalpy H	**Enthalpy Rise δH**	**Flow Q**	**$Q\delta H$**
Suphtr Outlet	1331			
4th Stg Fd Htr	345	986	1.000	986
H.P. Reheater	...	120	0.806	97
L.P. Reheater	...	145	0.734	107
Feed Pump	...	−5	1.000	−5
Summation				$\Sigma Q\delta H = 1185$

Overall Efficiency, E $= \dfrac{\Sigma Q\,\Delta H}{1.08} \times \dfrac{0.87}{\Sigma Q\delta H} = 0.272$

Oil Rate (18,500 Btu/lb) $= \dfrac{2545}{18{,}500E} = 0.505$ lb/shp/hr

Boiler Feed Rate $= \dfrac{2545 \times 1.08}{\Sigma Q\,\Delta H} = 6.86$ lb/sph/hr

Non-bleeding Turb. Water Rate $= \dfrac{2545 \times 1.08}{\Sigma\,\Delta H} = 5.26$ lb/sph/hr

APPENDIX

The following describes the method of calculating the turbine condition curves and the performance of the actual cycles.

Turbine Calculations

The turbine condition curve is drawn in convenient sections starting with the basic data from Table 2 and using the assumptions shown on Fig. 12. Intermediate pressures are chosen to give the desired division of work between turbines and to provide suitable bleeder and reheater points.

The pressure drop through the reheaters causes a loss of efficiency which has been taken account of in the reheat-cycle calculations. With a gas reheater the drop has been taken as 15 per cent of the reheater inlet pressure, based on experience with the "Examiner." The drop through steam reheaters has been taken as 3 per cent of the inlet pressure in the region of 200 psi abs and as 8 per cent in the region of 30 psi abs, based on designs that have been worked out in detail for an actual installation.

The internal efficiency of a section of the turbine between two pressure levels is the product of the basic stage efficiency, the moisture correction factor and the turbine reheat factor. The basic stage efficiencies for various pressures and the moisture correction factors are given on Fig. 12. Standard turbine reheat factors are used. The internal efficiency includes the effect of all internal conversion, moisture, rotation and leakage losses but does not include leaving, gear and bearing, and other external losses. The turbine leaving loss has been assumed 2 per cent, the gear and bearing loss 4 per cent and other external parasitic losses 2 per cent, the total external losses being 8 per cent.

The enthalpy ΔH extracted from each pound of steam in a section of the turbine is the product of the adiabatic enthalpy drop and the internal efficiency. These extractions are used to construct the condition curves. The sum of the extractions for all sections $\Sigma\,\Delta H$ divided by 1.08 is the external work performed by each pound of steam passing through the turbine when operating non-bleeding. Therefore the non-bleeding turbine water rate is:

$$\frac{2545 \times 1.08}{\Sigma\,\Delta H}$$

The water rates calculated by this method agree closely with trial measurements and therefore it is unimportant whether the actual losses are distributed exactly as assumed above.

Performance of Actual Cycles

The oil consumptions, water rates, etc., for the actual cycles have been calculated as follows: A line diagram of the cycle is drawn as shown on Fig. 13 and on it are entered the pressures, temperatures and enthalpies at all points shown. These data are obtained from the condition curve and from the following paragraphs.

The total quantity of feedwater supplied to the boiler is represented by 1.00 and all the other flows are expressed as fractions of this. All steam flows are entered on the diagram as soon as they are known. The steam quantity supplied to the auxiliary turbine-generator and for other auxiliary purposes is assumed to be 0.07 of the total. This is based on detailed estimates of the requirements for actual ships having all auxiliaries, except feed pumps, electrically driven, and it includes allowances for makeup feed, air ejector steam, galley steam, etc., see the next section for feed pump data.

The terminal temperature difference is assumed 10 deg F for the surface feed heaters and 0 deg F for the second-stage direct-contact heater. The second-stage heater pressure is taken as 10 psi g where bleeding is used to supplement the feed pump turbine exhaust, but it is allowed to build up a little higher in some cycles to absorb excess exhaust, there being no bleeding in such cases.

The bleeder steam required for each feed heater is calculated by means of a local heat balance for that heater. If the drains cascade from one feed heater to another, it is convenient to start with the highest pressure heater. The flow through each section of the turbine is calculated from these data.

The losses in the feed pump are converted into heat which increases the temperature of the feedwater as indicated on Fig. 13. Also, the useful work of the feed pump increases the enthalpy of the contents of the boiler drum as shown on Table 4. Thereby some of the feed-pump power is recovered.

Enter enthalpy and flow data in Table 4 and calculate $\Sigma Q\,\Delta H/1.08$ which is the net external work done by the turbine for each pound of boiler feedwater. Similarly calculate $\Sigma Q\delta H/0.87$, the total heat supplied by the fuel to the boiler and reheaters per pound of boiler feedwater. The boiler efficiency is assumed as 0.87 for all cycles. The overall efficiency, oil rate and water rates are calculated as shown on Table 4. Steam flows to the reheaters can be obtained by writing local heat balances but they do not enter directly into the performance calculation.

To find the gain due to feed heating repeat the above procedure with the bleeders shut off and compare with the efficiency already obtained.

Feed-Pump Loss Calculation

The power and steam requirements of the feed-pump turbine to be used in the plant performance

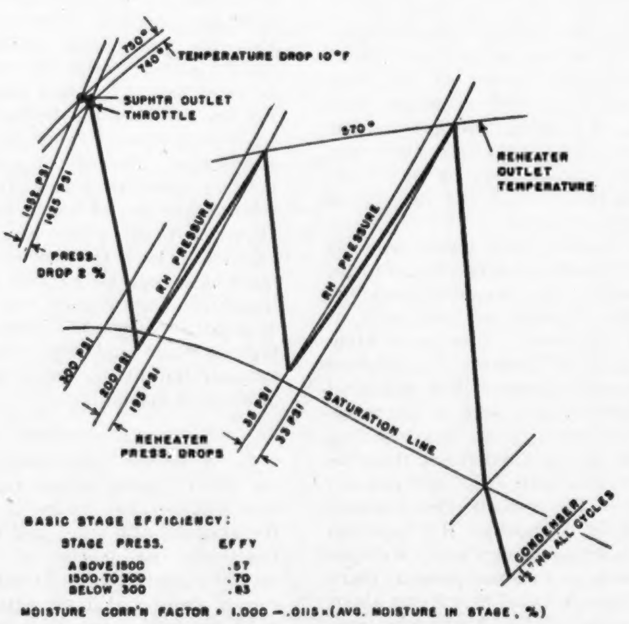


Fig. 12—Turbine condition curve for Cycle 7

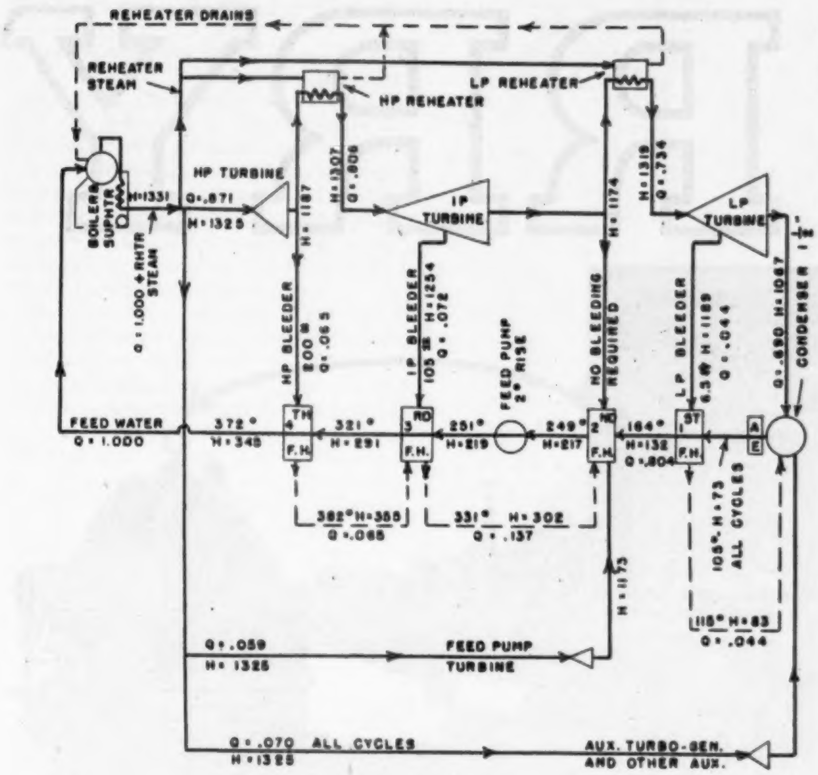


Fig. 13.—Flow diagram for Cycle 7

calculations and a close approximation to the feed-pump loss are derived below. As this loss is small it has been calculated by a method of differences.

- e = feed-pump efficiency
- E = overall efficiency of plant
- 0.87 = boiler efficiency
- f = ratio of total feed to non-bleeding turbine water rate
- p = feed-pump discharge pressure, psi
- q = non-bleeding main turbine water rate, lb/shp/hr
- q' = water rate of feed-pump turbine, lb/hp/hr
- r = replacement factor for bleeder serving 2nd stage heater
- v = specific volume of feedwater passing through pump, cu/ft/lb

All of the following equations have been written for one shaft horsepower output.

$$\text{Feed-pump turbine horsepower} = \frac{fqpv}{13{,}750e}$$

$$\text{Steam to feed-pump turbine } Q = \frac{fqq'pv}{13{,}750e}$$

When this quantity of exhaust steam Q is sent to the feed heater the amount of steam bled from the main turbine is reduced by hQ, where h is given by:

$$h = \frac{\text{Enthalpy of aux exh} - \text{enthalpy of liquid at bleeder press}}{\text{Enthalpy of bled stm} - \text{enthalpy of liquid at bleeder press}}$$

Therefore, less steam is needed at the turbine inlet to produce the same shaft horsepower, the reduction being rhQ. Then the net increase in steam from the boiler is $(1 - rh)Q$ and the loss due to it is:

$$\text{Loss due to feed pump steam} = \frac{(1 - rh)q'pv}{137.5e}\%$$

All of the water horsepower of the feed pump is supplied directly to the boiler drum as mechanical energy and is available to do work in the cycle. It is assumed that about two-thirds of the pump losses are internal and that these are imparted by friction to the feedwater as sensible heat. The addition of heat at this point of the feed cycle is less effective than if it were added directly to the boiler but the effect of this is negligible in most cases. Therefore, it may be assumed as an approximation that 87 per cent of the total feed-pump power is added to the boiler contents directly. This reduces the heat input required from the fuel and the reduction may be expressed as a percentage of the total input, $0.87/E$, as follows:

$$\text{Gain due to feed-pump power} = \frac{fqpvE}{137.5e}\ \%$$

The net feed-pump loss is the difference of these effects, namely,

$$\text{Net feed pump loss} = \frac{(1 - rh)q'pv}{137.5e}\left[1 - \frac{Efq}{(1 - rh)q'}\right]\%$$

The quantities in this formula have been varied systematically with pressure to find the variation of feed-pump loss. Based on data for actual installations the differential between the feed pump discharge pressure and the boiler pressure varies from about 120 psi in a 425-psi power plant to about 215 psi in a 1425-psi plant. It has been assumed that the excess pressure varies as the square root of the boiler working pressure between these two known points. The boiler drum pressure has been assumed 5 per cent higher than the throttle pressure.

The efficiency of the centrifugal feed pump has been taken as 0.76 for a 425-psi installation and 0.68 for 1425 psi. A suitable law of variation has been assumed between these two points.

The water rate of the steam turbine driving the feed pump has been taken as 20 lb per bhp per hr with 1425-psi 740-F steam and 10 psi g back pressure, based on guarantees obtained for actual installations. The turbine efficiency has been assumed substantially constant for all steam conditions and therefore the water rate is proportional to the adiabatic enthalpy drop from the turbine inlet steam conditions to 10 psi g.

The variations of the other quantities entering into this formula are relatively less important. The replacement factor r can be calculated from the condition curve for each cycle but its whole range of variation is only from 0.38 to 0.48. The specific volume v is 0.017 cu ft per lb, corresponding to about 250 F, the temperature of the feed-pump suction for all cycles.

The bracketed factor in the loss equation does not vary widely and may be taken as 0.83. However, if a more exact figure is desired the quantities necessary to calculate it are all available from the cycle performance calculations.

In order to establish the feed-pump turbine exhaust enthalpy for use in the performance calculations, the external losses of this turbine have been taken as 15 per cent.

This formula may also be used as the basis for calculating the feed-pump loss when electrically-driven positive-displacement pumps are used. In this case the pump efficiency may be taken as 0.85 and in place of q' substitute the turbine-generator water rate (converted to lb/hp/hr at the generator terminals) divided by the feed-pump motor efficiency, about 0.88. Usually the auxiliary turbine-generator exhausts to a vacuum and in this case r is zero. All of the other terms of the equation remain the same. In some cases the use of electrically driven feed pumps will necessitate the installation of a larger turbine-generator. Therefore, the entire electrical load will be obtained at a slightly better efficiency because the water rate of the larger unit is less. However, this gain is only of the order of 1/4 per cent on the oil rate and in some ships which have large generators for other reasons, there is no gain.

Acknowledgment

The author wishes to express his indebtedness to S. Curtis Powell, D.Eng., for his help in preparing this paper.

Progress Reported in Fuel Conservation Program

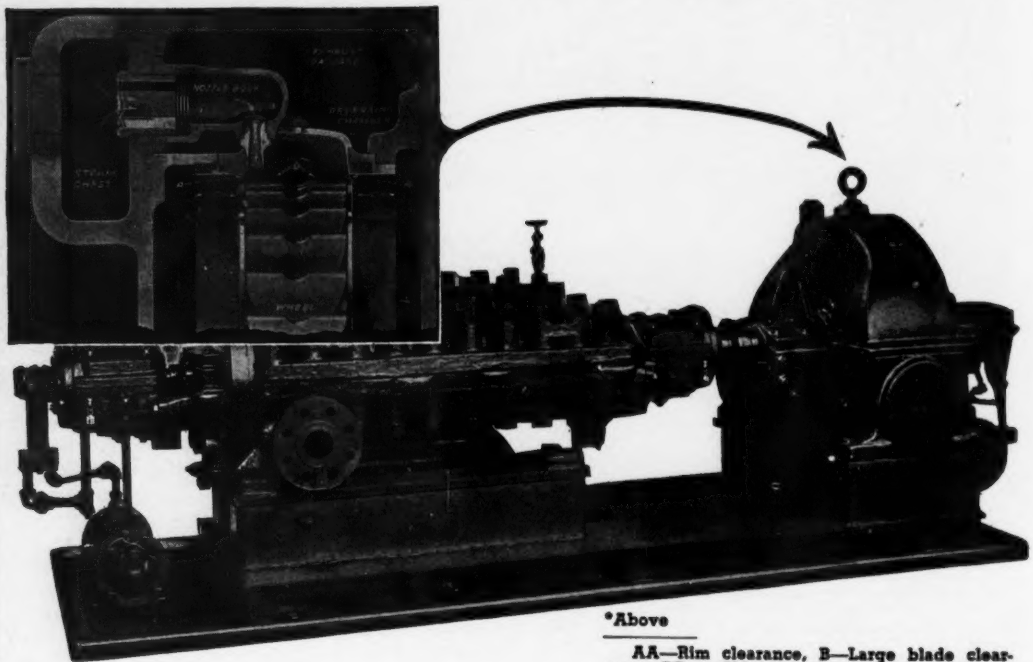
Secretary Ickes has announced the appointment of a district engineer and eight more coordinators as the Bureau of Mines prepare to carry its fuel efficiency program into the industrial and commercial plants in metropolitan areas within the next few weeks.

With thousands of volunteer regional engineers enrolled and a total of 12 coordinators named to direct their work in some of the principal industrial centers, organization of the campaign to minimize waste in use of coal, coke, wood, petroleum, and gas is well under way. The program, directed by the National Fuel Efficiency Council in cooperation with the Bureau of Mines, later will be expanded to include small fuel consumers.

John G. Mingle, of the Bureau of Mines, has been appointed district engineer, and assigned to Indianapolis, Ind., from where he will supervise the program in Michigan, Ohio, Kentucky, and Indiana. The new coordinators and their headquarters cities are M. F. Blankin, president, American Society of Heating and Ventilating Engineers, Philadelphia; E. L. Crosby, of the Henry Adams Co., Baltimore; P. W. Thompson, vice president Detroit Edison Co., Detroit; W. G. Christy, smoke abatement engineer, Hudson County Board, Jersey City; Sam Dalton, Dalton Coal and Material Co., Columbia, Mo.; F. K. Prosser, Norfolk & Western Railway Co., Roanoke, Va.; A. L. Maillard, Military Chemical Works, Inc., Pittsburgh, Kan.; and Linn Helander, head of the mechanical engineering department, Kansas State College, Manhattan, Kan.

Committees of engineers outstanding in the field of fuel utilization, representing each of the major fuel and allied industries, are preparing informational quiz sheets for use in the program. Pooling their technical information in the interest of conservation, these men will ask and answer questions that arise in obtaining maximum fuel efficiency. Armed with this information, together with that provided by the extensive research of Bureau of Mines employees, regional engineers soon will start visiting plants in their sections. Serving without pay, these engineers will request that the management of each company whose plant they visit sign a pledge to cooperate.

TERRY



*Above

AA—Rim clearance, B—Large blade clearance, CC — Side clearance, (about one inch). Blades cannot foul, as they are protected by rims. Rubbing at AA will do no damage. Side clearance is so large that end-play from excessive external thrust cannot damage wheel.

END PLAY WILL NOT DAMAGE THE BLADING OF THE TERRY WHEEL TURBINE

In the Terry Wheel Turbine the blades are protected by rims at the sides of the wheel, which would take without damage any rubbing that might occur if the clearance became reduced. With this construction it is impossible for the blades to foul and frequent inspection of the thrust bearing is not required to obtain safe and dependable operation.

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Unit Power Plants for China

In a paper before the American Section of the Chinese Institute of Engineers, published in the latest issue of that institute's *Journal*, U. K. Chan outlines the plans for providing China with a considerable number of unit power plants to supply lighting and power to industrial centers. Although negotiations for much of the equipment have been completed, actual shipments are awaiting the reopening of the main transportation routes.

Mr. Chan points out some of the factors governing the selection of this equipment. Chief among these were limitations of weight and size, for in most cases it would be necessary to transship the components by trucks to the ultimate destinations. The limitation on individual pieces was 5 tons gross and 12 ft in length. Moreover, in order to avoid the possibility of destruction by bombing, some of these plants may have to be set up in caves or tunnels, and it is important that their size and arrangement be such as to permit moving to other locations to avoid seizure through enemy advance. Finally, the question of availability of spare parts made standardized design desirable.

These considerations led to the adoption of 1000-kw and 2000-kw plants consisting of one boiler and one turbine-generator with complete self-contained accessory equipment capable of operation independent of other units. Where greater capacity is required it is planned to use two or more such units with interconnection on the steam ends.

Briefly, the steam-generating unit involves a bent-tube boiler with superheater, economizer or air preheater, water walls and soot blowers, forced- and induced-draft fans and control equipment.

The boiler, which for the 1000-kw plant is designed for 21,000 lb per hr continuous output, will be fired with a spreader stoker combined with a traveling grate. There will be a dual-drive feed pump, condensate pump, deaerating heater, evaporator, tanks, miscellaneous pumps, etc. The turbine is a seven-stage, 5500-rpm machine driving the 1000-rpm three-phase, 6900-volt generator through a reduction gear. Use of a geared unit was due to the necessity of keeping down shipping weights. It will run condensing and be provided with a two-stage air ejector. A complete switch gear is provided as well as all the necessary piping and valves.

The steam conditions selected are 425 psi at the superheater outlet and 740 F total steam temperature. This corresponds to 400 psi 725 F at the turbine throttle and operation with 28-in. vacuum is planned. The calculated plant heat rate is approximately 21,400 Btu per net kilowatt-hour output with an anticipated boiler efficiency of 80 per cent.

Choice of these steam conditions was dictated by the employment of standard valves and fittings and the expectance of reasonably high efficiency, also with a view to post-war expansion.

Selection of spreader stokers was influenced by their ability to burn a wide variety of fuels for, due to the difficulties of transportation in China, most of these plants will have to burn whatever coal is available in the vicinity.

Midwest Power Conference Scheduled for April 13–14

The Seventh Annual Midwest Power Conference, arranged by Illinois Institute of Technology, will be held on April 13 and 14 at the Palmer House, Chicago. The program will deal with both war and post-war problems and will comprise nine sessions, namely, the opening general meeting, three eleotrical sessions, one on power plant practice, one on industrial power plants, another on plant maintenance, and sessions on fuels and on diesel power.

The keynote speaker will be Alex D. Bailey of the Commonwealth Edison Company whose topic will be "Post-War Planning of the Nation's Power Supply"; and J. A. Krug, director of the Office of War Utilities of the War Production Board will be the principal speaker at the All-Engineers' Dinner.

According to Prof. Stanton E. Winston, director of the Conference, approximately twenty-five nationally known power authorities will appear on the program. As in former years, the American Society of Mechanical Engineers and the American Institute of Electrical Engineers will each sponsor a conference luncheon.

Coöperating with the Illinois Institute of Technology in sponsoring the Conference, are Iowa State College, Michigan State College, Northwestern, Purdue, Iowa, Illinois, Michigan and Minnesota Universities and eight of the engineering societies.

The preliminary program as of February first is as follows:

Thursday April 13, 1944

9:00 a.m. Registration, Palmer House.

10:15 a.m. Opening Meeting. H. B. Dirks, Chairman.

(*a*) Address of Welcome. Robert B. Harper, Vice President, Peoples Gas Light and Coke Company, Chicago.

(*b*) Response for the Coöperating Institutions. F. M. Dawson, Dean, College of Engineering, State University of Iowa.

(*c*) "Post-War Planning of the Nation's Power Supply," Alex D. Bailey, Assistant to the Vice President, Commonwealth Edison Company, Chicago.

12:15 p.m. Joint Luncheon with A.S. M.E. J. P. Magos, Chairman.

Speaker: C. O. Dohrenwend, Chairman, Engineering Mechanics Research, Armour Research Foundation, Chicago. "Review of Experimental Stress Analysis Methods."

2:00 p.m. Central Station Practice. R. K. Behr, Chairman. (Sponsored and arranged by the Power and Fuels Division, Chicago Section, A.S.M.E.)

(*a*) "Progress Towards Standardization of Turbine-Generators for Central-Station Service," M. S. Oldacre, Equipment and Research Engineer, Commonwealth Edison Company, Chicago.

(*b*) "Causes and Prevention of Boiler Priming and Solids Carryover," W. H. Rowand, Engineer, Babcock & Wilcox Company, New York.

(*c*) Discussion.

3:45 p.m. Plant Maintenance. H. L. Solberg, Chairman.

(*a*) "Maintenance Systems," W. A. Perry, Superintendent, Electric and Power Departments, Inland Steel Company, East Chicago, Indiana.

(*b*) "Corrosion Problems," L. G. Vande Bogart, Research Engineer, Research and Development Laboratories, Crane Company, Chicago.

(*c*) Discussion.

3:45 p.m. Electrical Session No. 1. J. E. Hobson, Chairman.

(*a*) "Studies of Generator Armature Insulation Deterioration," F. J. Pohnan, Senior Testing Engineer, Commonwealth Edison Company, Chicago.

(*b*) "Temperature Deterioration of Solid Insulation in Oil," F. J. Vogel, Professor of Electrical Engineering, Illinois Institute of Technology.

(*c*) Discussion.

6:45 p.m. "All Engineers" Dinner. Informal. Grand Ball Room.

Toastmaster: H. B. Gear, Vice President, Commonwealth Edison Company, Chicago.

Speaker: J. A. Krug, Director of War Utilities, War Production Board, Washington, D. C.

Friday, April 14, 1944

9:00 a.m. Industrial Power Plants. Hugh E. Keeler, Chairman.

(*a*) "Application of Electronics to Power Generation," J. D. Ryder, Assistant Professor of Electrical Engineering, Iowa State College.

(*b*) "Inhibitors in Condenser Tube Alloys," Austin R. Zender, General Sales Director, Bridgeport Brass Company, Bridgeport, Connecticut.

(*c*) Discussion.

10:30 a.m. Fuels. R. E. Summers, Chairman.

(*a*) "Research in Fuels as a Postwar Necessity," A. R. Mumford, Development and Research Department, Combustion Engineering Company, New York.

(*b*) "Postwar Outlook for Oil Fuels," Arch L. Foster, Refinery Editor, *The Oil and Gas Journal*, Tulsa, Oklahoma.

(*c*) "Statistical Comparison of Fuels," C. C. DeWitt, Chairman, Department of Chemical and Metallurgical Engineering, Michigan State College.

(*d*) Discussion.

10:30 a.m. Electrical Session No. 2. R. W. Jones, Chairman.

(*a*) "Developments in the Field of Control", E. H. Alexander, Engineer, Industrial Control Division, General Electric Company.

(*b*) "New Applications of Mercury Arc Rectifiers," W. E. Gutzwiller, Engineer in charge of Rectifier Application, Allis-Chalmers Manufacturing Company, Milwaukee.

(*c*) Discussion.

12:15 p.m. Joint Luncheon with A.I.E.E. F. E. Keith, Chairman.

Speaker: B. W. Clark, Vice President in charge of Sales, Westinghouse Electric and Manufacturing Company, East Pittsburgh. "Tomorrow's Outlook."

2:00 p.m. Electrical Session No. 3. L. B. Les Vesconte, Chairman. (Sponsored and arranged by the Power Group, Chicago Section, A.I. E.E.)

(*a*) "Trends in Power System Analysis," Miss Edith Clarke, Central Station Engineer, General Electric Company, Schenectady.

(*b*) "Ice Melting Practice on 132-kv Lines," H. A. Cornelius, Senior Engineer, Public Service Company of Northern Illinois, Chicago.

(*c*) "Visualizing Heat, Magnetic, and Electrostatic Field Problems in Electrical Apparatus," J. F. Calvert, Chairman, Department of Electrical Engineering, Northwestern University.

(*d*) Discussion.

3:45 p.m. Diesel Power. M. P. Cleghorn, Chairman.

(*a*) "Closed Evaporative Coolers," William Goodman, Consulting Engineer, Trane Company, La Crosse, Wisconsin.

(*b*) "Isolation of Vibration From Diesel Plants," H. H. Fink, Product Design Engineer, The B. F. Goodrich Company, Akron, Ohio.

(*c*) Discussion.

NEW CATALOGS AND BULLETINS

Any of these publications will be sent on request

A.S.T.M. Standards

The "List of A.S.T.M. Standards and Tentative Standards," dated October 1943, may be obtained from the American Society for Testing Materials at 25 cents each. The listing covers Specifications, Methods of Testing, Recommended Practices, Definitions of Terms, Charts and Tables. A supplementary insert lists A.S.T.M. Emergency Standards and Emergency Alternate Provisions issued up to October 31, 1943.

Carbon Seal Rings

An 8-page booklet, dealing with carbon seal rings and their application to the bellows type shaft seals, has been issued by the Pure Carbon Company. The bulletin is illustrated and drawings of seal designs are given.

Fire Brick

M. D. Valentine & Bro. Company has issued a 16-page catalog covering its line of firebrick and specialty products. Raw material resources and manufacturing facilities are described and illustrated. Other sections deal with special shapes, Star 1-A firebrick, mortars and clays, and Valcloset high-temperature cement. The catalog is profusely illustrated and four pages are devoted to standard shapes and firebrick information.

Heat Exchangers

The Brown Fintube Company has issued a 6-page folder describing its "Sectional Hairpin" heat exchangers. This bulletin (No. 432) is admirably illustrated and includes dimensional data of standard sizes, and also performance data of four different exchangers.

Heat Enclosures

George P. Reintjes Company has issued a 4-page bulletin (No. 431) "To Conserve Fuel" which illustrates and describes the special features of its sectionally supported wall construction, for the upper side wall areas of bent-tube boilers, which permits freedom of expansion of the drums while maintaining a tight air seal around the drum ends.

Pumps

"How to Pump It" is the title of an attractive 20-page bulletin (No. 433) just issued by Chain Belt Company and describing its line of Rex Speed Prime Pumps. Complete information concerning construction and operation of these pumps is given together with capacity charts and specifications. Pump capacities range from 2000 to 125,000 gallons per hour.

pH and Chlorine Control

The 6th edition of the W. A. Taylor & Company combination handbook and catalog has just been issued. This 83-page publication has been completely revised. Fifty pages are devoted to a simple explanation of the meaning of pH value and the methods of making colorimetric determinations, the application of pH and chlorine control, the application to boiler water (containing sections on water purification and softening, phosphates, silica and steam condensate) and a technical discussion of the meaning of pH control. The remaining pages describe Taylor Slide Comparators for general pH and chlorine control, and for determination of phosphates, silica, total iron, etc., in boiler water.

Surplus Steel Valves

A 50-page catalog of 30,690 new surplus steel valves has been published by the Surplus Program Section of the WPB Redistribution Division for the use of war contractors or others. It indicates the type of valves, catalog specifications, manufacturers' names and state location. It will be available for inspection by steel valve users in WPB's Regional or District Offices. Claimant Agencies will also have copies of the catalog available.

All war contractors or other persons who have need for steel valves for war use are urged to get in touch with the nearest WPB field office, in order to determine if the types of valves needed are available. Persons needing steel valves may learn their location through the Surplus Program Section of WPB, 155 East 44th Street, New York City.

Tube Cleaners

Bulletin Y-15, a 4-page publication issued by the Elliott Company, describes the new and improved Elliott 1300 Series tube cleaner. Motors have been proportioned so that they can be used to clean both straight and curved tubes, with parts interchangeable with former Lagonda Cleaner motors. Also described is the new FH coupling, which is described in more complete detail in supplementary Bulletin Y-17.

Pulverizer Manufacturers Confer With Bituminous Coal Research, Inc.

At the invitation of the Committee on Industrial Utilization of the Technical Advisory Board of Bituminous Coal Research, Inc., representatives of the manufacturers of pulverized-coal equipment met with the Committee at Battelle Memorial Institute in Columbus, Ohio, on January 12 to discuss ways and means by which research could advance the use of pulverized coal. Vernon G. Leach, Chairman of the Committee, and Julian E. Tobey, Chairman of the Technical Advisory Board of B.C.R.I., explained that the research agency had for several years been carrying on research in a modest manner on the application of pulverized coal to forge furnaces and to radiant-tube furnaces. With the greatly expanded program planned by B.C.R.I. which is expected to reach a rate of expenditure of $500,000 annually, the opportunity will be offered for the initiation of new projects. The advice and counsel of the manufacturers was wanted to insure that the needed information was obtained.

Prompted by an agenda that had been prepared to initiate discussion, the conferees considered the need for an abrasion index for coal and for coal ash, an abrasion index for metals relative to coal or ash, the relation of surface and inherent moisture in coal to its grindability, the mechanism of combustion, the rate of flame propagation, the fundamentals of the flow of coal and air in pipes, the effect of recirculation of flue gases on combustion, the fusion of coal ash and radiation from flames.

The B.C.R.I. Committee stated that it was definitely interested in projects that would lead to improved performance of coal fired under stationary steam boilers, but felt that the technology and engineering of that field were well in hand. Its particular interest was in extending pulverized coal into the non-steam uses where oil and natural gas had so largely supplanted coal. Discussion on these uses covered the open-hearth furnace, cement, lime, and brick kilns, railroad locomotives and the gas turbine.

The firing of an experimental furnace fitted with four alloy steel radiant tubes with pulverized coal was demonstrated to the group in the Battelle laboratories. The pulverized coal is distributed to the four tubes by a circulating system. Ignition is effected automatically by electric spark and gas pilot. More uniform temperature distribution along the tubes and higher rates of heat input with uniformity of temperature were reported to be possible with coal than with gas firing. The confining of the ash of the coal within the tube from which it is swept by the high-velocity gas stream offers many new opportunities for coal.

To continue the cooperation and to work up a definite program from the many good suggestions, a permanent committee from the manufacturers was named, consisting of Ollison Craig, J. E. Crites and R. M. Hardgrove.

The representatives of the manufacturers in attendance were: Babcock & Wilcox Co., R. M. Hardgrove and L. S. Wilcoxson; Combustion Engineering Co., John Van Brunt; Foster Wheeler Corp., John Blizard; Kennedy-Van Saun Engineering and Manufacturing Co., A. R. Allen; Riley Stoker Co., Ollison Craig; Sims Co., W. A. Pettibone; and Whiting Corp., A. J. Grindle. The personnel of the Industrial Utilization Committee of B.C.R.I. are V. G. Leach, Peabody Coal Co., Chairman; E. C. Payne, Consolidation Coal Co.; C. J. Potter, Rochester and Pittsburgh Coal Co.; J. C. Scott, New River Co.; L. A. Shipman, Southern Coal and Coke Co.; R. F. Stilwell, Red Jacket Coal Sales Co.; M. A. Tuttle, Knox Consolidated Coal Co.

Members of the staff of the Coal Research Laboratory of the Carnegie Institute of Technology present were H. H. Lowry, Director, and A. A. Orning. Members of the Battelle staff were: R. A. Sherman, Supervisor, and B. A. Landry, Assistant Supervisor, Fuels Division, and W. H. Browne, R. B. Engdahl and E. R. Kaiser, Research Engineers.

Coal Stocks Continue to Decline

Concern is being felt by the Solid Fuels Administration over the fact that bituminous coal stocks in the hands of consumers had dropped to an average of 30 days' supply as of January first, and they are probably considerably lower by now.

During December when the consumption of soft coal increased 15.7 per cent over the November figure, consumers used 3,387,000 tons more than was mined. While it is not unusual for consumption to exceed production during the mid-winter months when fuel requirements are at their peak, the fact that stocks have fallen to such a low point in the face of increasing war requirements presents a serious problem.

While the electric utilities averaged two months' supply on hand, steel mills were down to three weeks and many industrials are now getting their coal on a hand-to-mouth basis. It has therefore been necessary to cause plants with comparatively large stockpiles to reduce their orders for current shipments so that coal can be diverted to other consumers less adequately supplied. This has resulted in some plants being required to cut their current orders to as little as 50 per cent and make up the difference by drawing upon their storage piles.

Despite this situation the bituminous coal production is slowly climbing back toward previous high levels and that for the week ending January 29 was 12,830,000 tons. The total January output showed an increase of 9.7 per cent over that of the same period last year. This, however, is less than the increased rate of consumption.

Following is the "Stock Limitation Table" prescribed by the Solid Fuels Administration to fix the maximum percentage of monthly consumption requirements that may be ordered by industrial and utility consumers in Districts 1 to 13, inclusive (excepting 5 and 12):

	Per Cent of Monthly Requirements			
	Utilities		Industrials	
Days' Supply	A*	B†	A*	B†
Under 15	140	140	120	120
15 to 20	130	130	110	110
21 to 25	120	120	100	110
26 to 35	110	110	70	110
36 to 40	100	110	70	100
41 to 55	70	100	70	70
56 to 69	50	70	50	70
70 or over	50	50	50	50

* Applicable to purchasers of bituminous coal shipped from any mine, yard or dock via any method of transportation to any destination except as specified in footnote†. A public utility having more than 36 days' supply may order such additional coal as is necessary to maintain 36 days' supply, and industrials with more than 21 days' supply may order coal in amount to maintain 21 days' supply.

† The percentages in this column are applicable to purchasers of bituminous coal shipped to Canada and of coal directly shipped via tidewater to any destination in New York Harbor or New England. A public utility in this group having more than 51 days' supply may order additional coal to maintain 51 days' supply, and an industrial with more than 36 days' supply may order additional coal to maintain 36 days' supply.

Peak Demand Up 12.5 Per Cent

The Federal Power Commission, in a report issued February 7, states that December 1943 peak demands of the principal electric utility systems of the country aggregated 37,063,961 kw, which was the highest monthly total for the year and 12.5 per cent above the peak of December 1942. The electric energy requirements for the month of December 1943 amounted to 19,662,813,000 kwhr which was a gain of 14.5 per cent over the corresponding period of 1942. The total energy requirements for the year 1943 reached 215,085,881,000 kwhr which was 17.2 per cent more than the 1942 total.

Incidentally, Class I electric utilities have estimated the total 1944 energy requirements as 229,565,239,000 kwhr, or an increase of 6.7 per cent over that of 1943. Their estimate for peak demand was an increase of 4.7 per cent. To meet this there are scheduled net additions to installed generating capacity of 1,696,750 kw for the current year. These figures represent only those additions actually on order and authorized by the War Production Board.

Personals

C. Davis Blackwelder, for 21 years associated with J. E. Sirrine & Co., consulting engineers of Greenville, S. C., has lately become vice president in charge of engineering for the Reynolds Metals Co. of Richmond, Va.

R. W. Owens has been appointed assistant to the president of Elliott Company, Jeannette, Pa. A graduate of the University of Illinois, Mr. Owens comes to Elliott Company from the Westinghouse Electric & Mfg. Company where he held various positions of responsibility for the past 28 years.

CORRECTION—Attention is called to a typographical error on page 51 and in the Table of Contents of the January issue. The author of the address on "What Does 1944 Hold for Business" was W. L. Batt and not W. L. Bott as printed.

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